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POWER

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Again the Coal Question

WHETHER or not to buy coal, or to contract for it, is going to be a serious question indeed during the coming month. The acuteness of the situation is particularly noticeable at this season, when it is customary to make new contract provisions for the supply of the new coal year. If the British miners strike, as is now freely prophesied, and thus export trade becomes an important factor in the market at an early date, there is every opportunity for the American coal-buying public to suffer again from stampede and threatened conditions which are commonly believed to represent shortage or panic.

If the American coal buyer begins to purchase calmly and carefully during the months of April and May, the demonstrated mine and railway capacity of this country will preclude any possibility of serious coal shortage. Nothing short of extreme catastrophe could prevent adequate supply before bad weather of next fall and winter if the orders begin to flow into the hands of coal producers at an early date. Exorbitant prices or a stampede of purchasers can find nc justification. If such unfortunate circumstance should occur as an effort to stampede buyers into purchase of excessive quantities or competitive bidding for coal supplies, everyone can well bear certain facts in mind. The United States can easily mine, haul and deliver eleven or eleven and one-half million tons of coal each and every week, barring unusual strikes in mine or on the railway system. For a few weeks it can even reach thirteen million tons a week and not overstrain our transportation and distribution facilities. This is much more coal than America can burn or even can burn and export also. Let the buyer well remember this fact and refuse to be stampeded into a runaway market, such as has been prophesied for late spring.

On the other hand, the buyer must not under any circumstances think that he can safely wait until next fall to lay in his stocks of coal. To the extent to which the buyer is today "coasting" on his last winter's stocks and waiting for prices to get lower, to that extent he is running a serious risk. It is well that stockpiles be cut down to a reasonable and proper minimum this spring before coal for the coming winter is placed in storage; but to wait much longer for material reduction in prices will be the most certain way in which to raise prices to unreasonable figures during the late spring and summer.

Whether to buy now or to contract for the next year's supply immediately, is of course a question for each executive to decide for himself after careful study of the credit conditions of his company, the certain needs for the coming year, the facilities for storage of coal and other pertinent factors. But for all alike there is one appropriate word of warning. With the first evidences of rising prices for coal, and these are already to be seen in places, do not rush in and help spread the panic of buying. At no time will sane, careful purchasing skill be more needed than in the near future. It will be most unfortunate if the American coal buyer forgets himself so far as again to bid wildly for fuel supplies at a time when we know that we have the coal, we have the miners, we have the coal cars, and that miner, mine owner and railroad will gladly work to supply the market at a *reasonable* margin. There is no excuse for offering the *excessive* margin which surely will be demanded if a panic of buyers ensues.

A Mine-Mouth Super-Power Station

SIX elements are essential to an industrial center; namely, raw material, water, fuel, transportation, labor, and proximity to a market. In these respects Pittsburgh is especially favored with regard to the iron and steel industry. As a result the Pittsburgh district is the largest steel-producing center in this country, and allied industries have been attracted to locations in or near the city.

As seen from Fig. 1, the Allegheny and Monongahela Rivers meet in the City of Pittsburgh and form the Ohio. These rivers furnish an abundance of water for manufacturing purposes, and proximity to coal mines makes the location ideal for steam-electric power production. For miles on either side of these rivers extensive industries have grown up, creating a large demand for electrical power, which can be satisfied only from a comprehensive and reliable source of supply.

In the Pittsburgh district, as in most other industrial centers, the early power development took the form of a number of small interests operating independent systems. By 1912 these had proved inadequate to meet the requirements of the rapidly growing industries and in an effort to provide an ample power supply for the future, these interests were consolidated to form the Duquesne Light Company. As a result, a comprehensive transmission system has been built in the form of a duplicate ring, Fig. 1, known as the "Duquesne Ring," around the industrial district of Pittsburgh. From this ring Beaver and Allegheny Counties, an area comprising 1,154 square miles with a population of 1,300,000, are supplied with power.

Brunot's Island power station may be considered the foundation of the Duquesne Ring, as it was in operation in 1912 and has since been enlarged until it now has a capacity of 120,000 kw. There are also six other plants located at different points on the system, ranging in capacity from 2,000 to 17,000 kw., making a total generating capacity available from all sources of approximately 160,000 kw. Power is delivered into the ring at 66,000 volts and distributed from the substations, indicated in the figures, at 22,000 and 11,000 volts over radial feeders into the industrial district of Pittsburgh, where it is again transformed for the various uses for which it is required.

Within the Pittsburgh district the growth of power demand has been rapid and within recent years has averaged about 15 per cent annually. In 1898 the connected load of the Duquesne Light Company's system was 8,000 kw. This had increased to 315,768 kw., as of Oct. 1, 1920, with a kilowatt-hour output of 805,000,000 for the

A DESCRIPTION of the engineering features and main generating units of the Colfax Power Station of the Duquesne Light Company. Colfax station is located near the company's coal mine, about fifteen miles from Pittsburgh, and will ultimately have 300,000-kw. capacity. At present one 60,000-kw. unit is in operation and a second one is on order. Power generated at 12,000 volts is stepped up to 66,000 volts through a bank of three 23,600-kva. transformers in the power house and is delivered through high-tension buses on the roof to an outdoor switching station supplying transmission lines. Each boiler has 20,876 square feet of heating surface. Boilers are fired under the low end. Although the station was built when material and labor were at a maximum price, the cost of installation has been kept within $105 per kilowatt. A very simple and convenient layout has been obtained, designed to give maximum economy and reliability.

year at a load factor of 57 per cent. This rapidly increasing demand made necessary some provision for extending the system along the most economical lines and as a result a site for a new power station was chosen at Colfax, on the Allegheny River. It is evident from Fig. 1 that this is a logical location, being on the opposite side of the ring from the Brunot's Island plant. From an analysis of load growth it was found that, in the Pittsburgh district for some years to come, the greatest increase in demand will be in the direction of Colfax, which is only about 15 miles from the center of Pittsburgh, and three-quarters of a mile from the Harwick mine of the Equitable Coke Co., an affiliated interest of the Duquesne Light Company. At this location the minimum flow of the Allegheny River is sufficient to supply circulating water to develop 300,000 kw., and this has fixed the maximum capacity of the development at this value. However, this limit is so high as to make this location an ideal one for a mine-mouth power plant. Units of 60,000-kw. rating are being installed, and with one spare, the ultimate capacity of the station is planned for 300,000 kw. The power station has been laid out so that it can be extended until all the available condensing water is utilized. Thus in the ultimate development there will be six units, and when the plant is completed it will not be complicated with a large number of small machines. Three buildings constitute the plant—a boiler room, a turbine room and an electrical bay, with only division walls between them (see Fig. 9) running parallel with the river. This has greatly simplified the design and also rendered it possible to make the initial installation

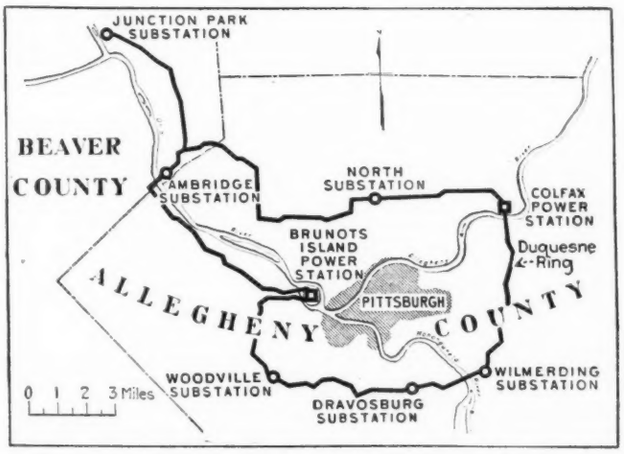


FIG. 1. DUQUESNE RING, A 66,000-VOLT LOOP 89 MILES IN CIRCUMFERENCE

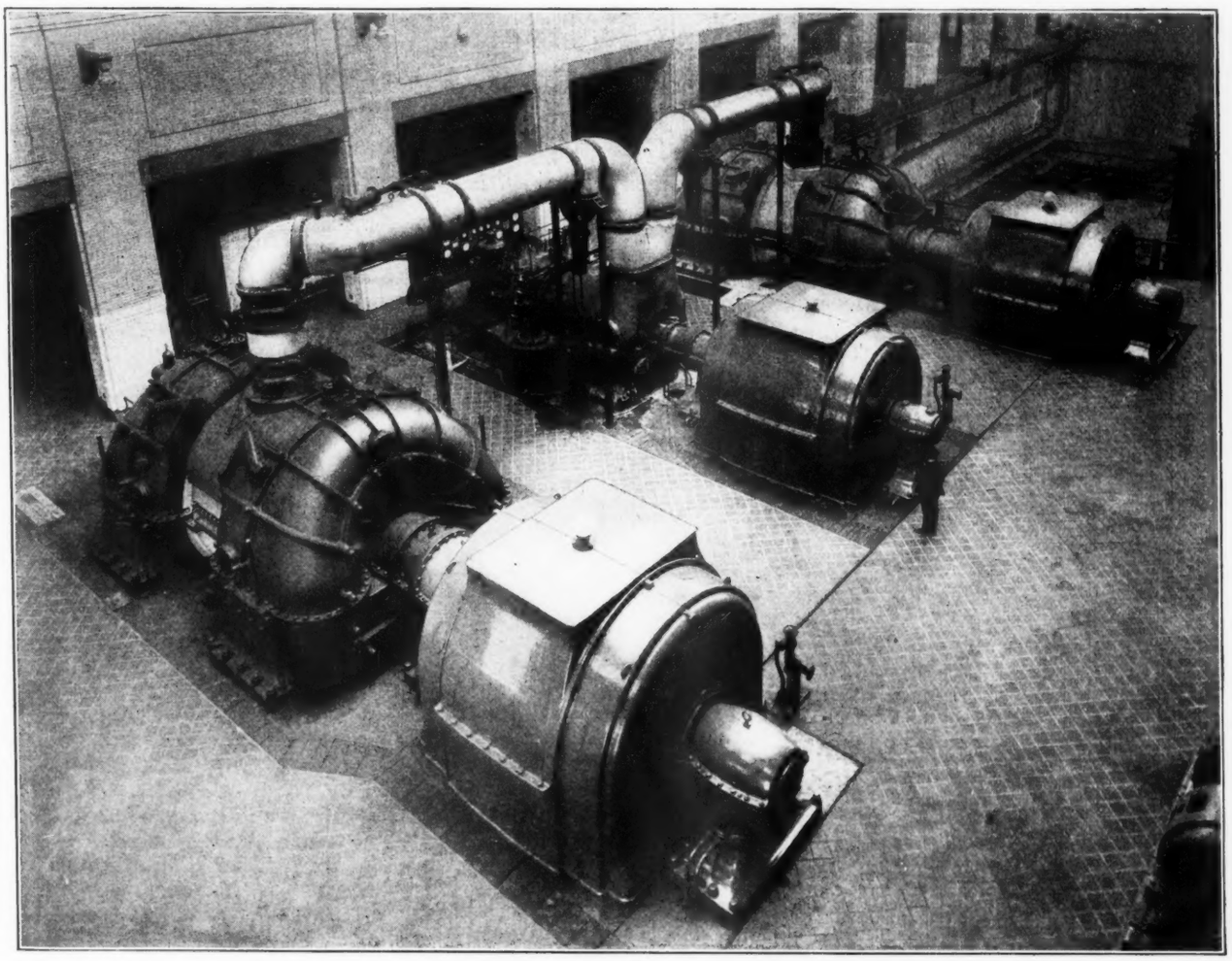


FIG. 2. TURBINE ROOM, SHOWING 60,000-KW. UNIT; HIGH-PRESSURE ELEMENT IN CENTER

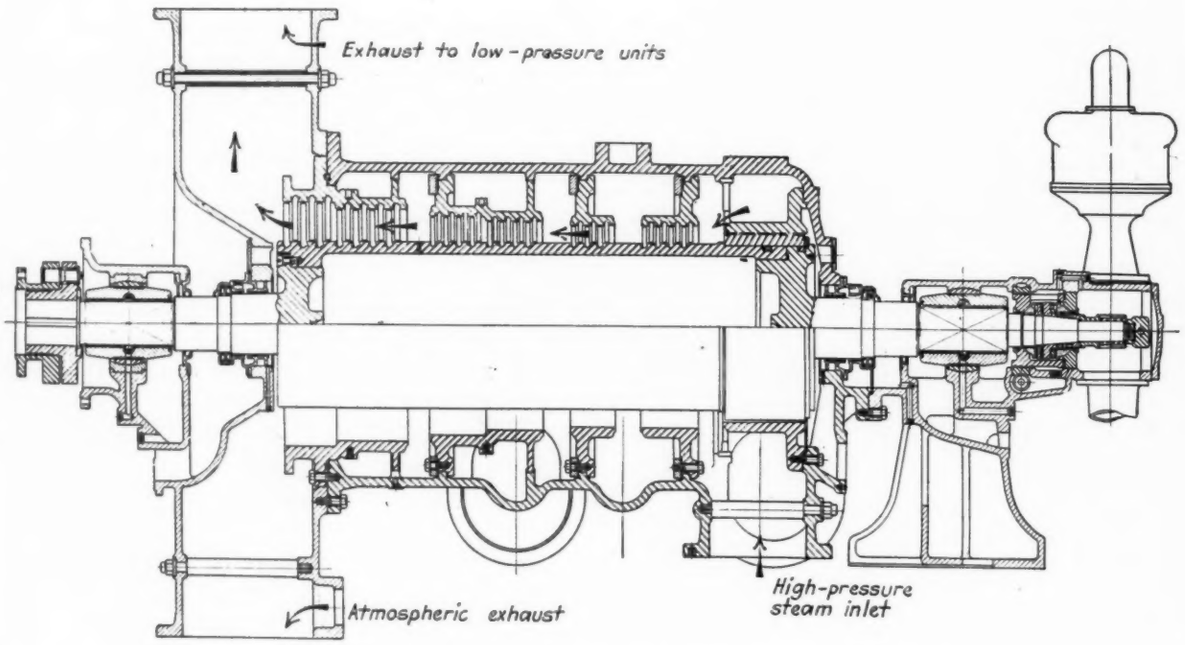


FIG. 3. SECTIONAL VIEW THROUGH THE HIGH-PRESSURE TURBINE

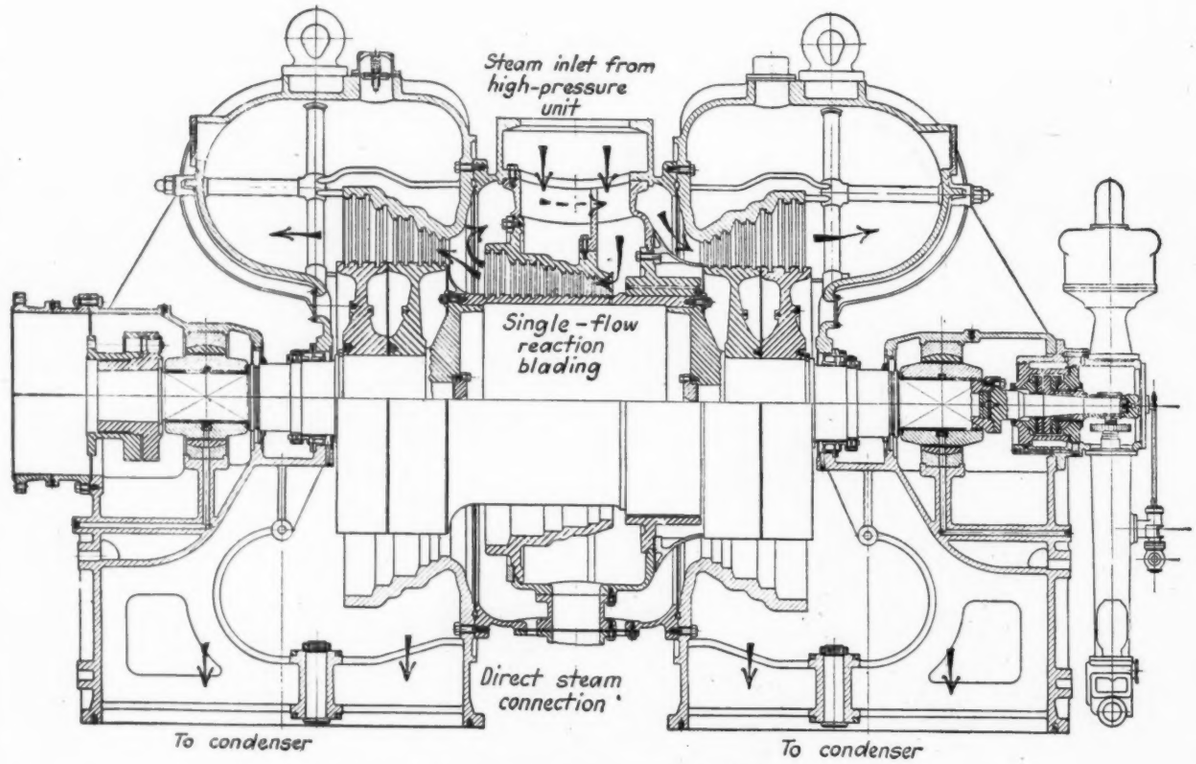


FIG. 4. SECTIONAL VIEW THROUGH ONE LOW-PRESSURE TURBINE

The high-pressure rotor of Fig. 3 is built up of three main pieces of high-grade steel—a central body or drum and two shaft ends. Both end sections are secured to the drum by long spigoted press fits, reinforced at the coupling end by tap bolts and at the governor end by shrink links. The low-pressure rotor of Fig. 4 is built up of a central section, consisting of a hollow drum on which are mounted the single-flow reaction blading and labyrinth packing. The shaft ends are secured to this drum, and each carries two discs of low-pressure reaction blading. Steam is supplied to the high-pressure element at 265 lb. gage and 175 deg. superheat, is expanded down to about 55 lb. at full load, and is carried by overhead pipes to two low-pressure elements which expand down to about one inch of mercury.

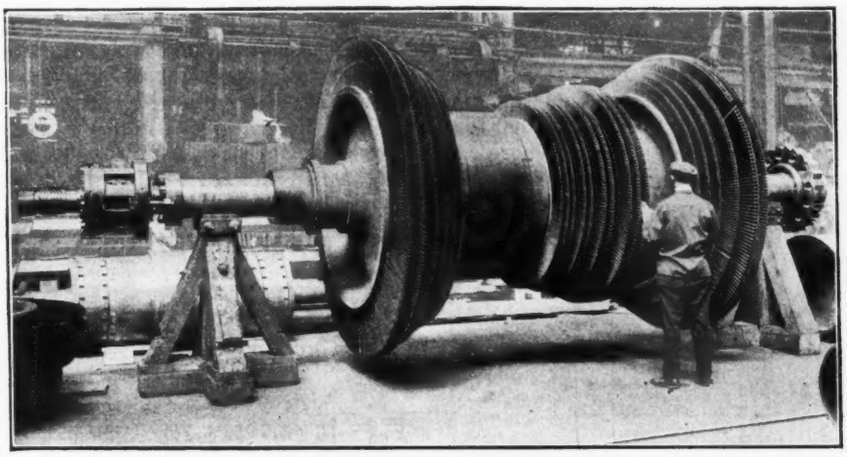


Fig. 5. Rotor for one of the low-pressure elements. Note the single-flow reaction blading in the right center, through which the steam passes before going to the double-flow blading at each end. A labyrinth packing is placed between the low-pressure blading on the left and the single-flow blading in the right center. The rotors of the two low-pressure elements operate at 1,200 r.p.m. and that of the high-pressure element at 1,800 r.p.m.

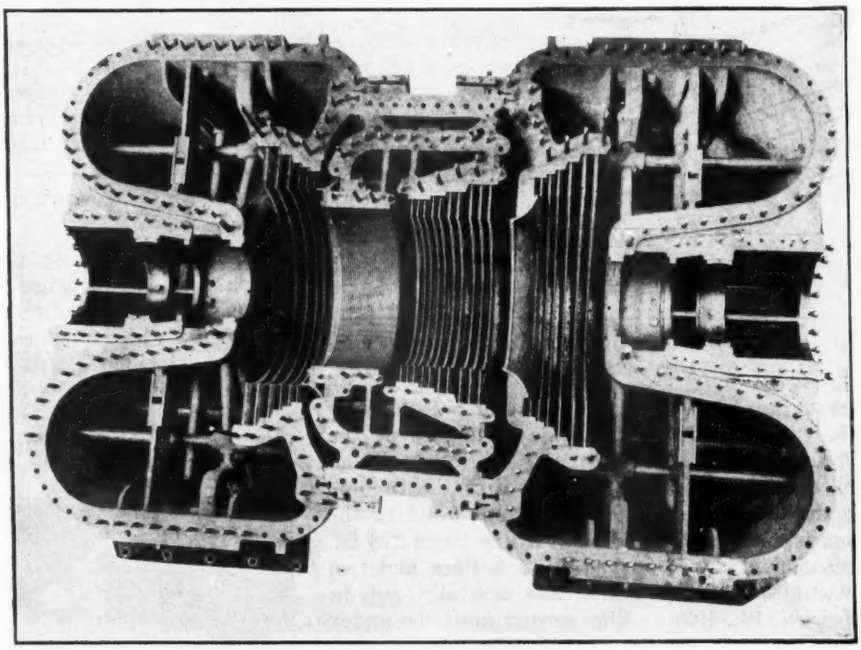


Fig. 6. View looking down into the bottom half of the casing for one low-pressure unit. Not only is the arrangement of the blading shown, but also the bracing in the exhaust chambers. The length of blading in the high-pressure elements ranges from 4 to 9 inches and in the low-pressure elements 4 to 18 inches. At full load of 60,000 kilowatts 648,000 pounds of steam per hour will pass through the three cylinders. This steam at the inlet to the high-pressure element occupies a space of about two cubic feet per pound and at discharge from the low-pressure elements has a volume of approximately 600 cubic feet per pound.

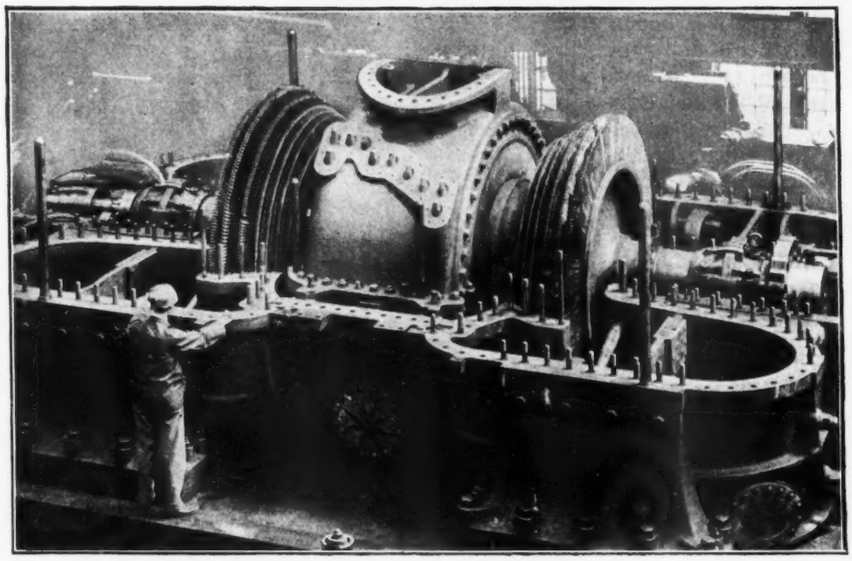


Fig. 7. Lower half of casing with rotor in place. The stationary single-flow reaction blading is carried in a cast-steel section, independent from the main casing, also shown in place in the center of the figure. The central portion of the main cylinder is also made of steel, so that in event of operation with steam direct from the boiler, all parts subjected to high temperature are of steel. When the low-pressure elements are operating independently, they will develop 30,000 kilowatts on a steam consumption of 13.8 pounds per kilowatt-hour.

at a minimum cost. One of the major considerations in the design of this plant was to produce an installation that would generate power at the lowest unit cost permitted by the state of power-plant development, including fixed charges.

The plant has an actual width of 240 ft. over all, made up of 108 ft. for the boiler room, 84 ft. for the turbine room and 48 ft. for the electrical bay. The

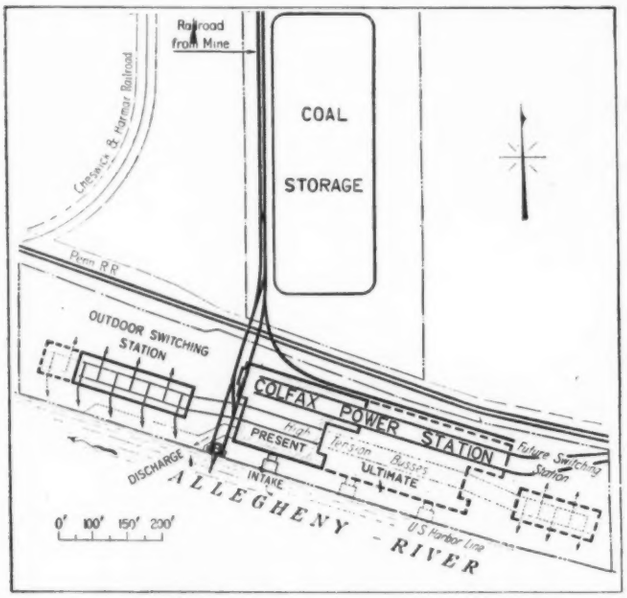


FIG. 8. LAYOUT OF POWER-STATION SITE

boilers are arranged in two rows running parallel with the turbine room, and in this design the size of the boilers is such that it has been possible to obtain an approximate balance between the length of the turbine room and the boiler room. In the present building the boiler room is 350 ft. long and the turbine room 278 ft., affording sufficient space to install 14 boilers and two 60,000-kw. units, one of which was officially put in operation Dec. 18, 1920. The second unit is under construction and will be installed in the near future. Actual design and construction of the plant was begun by Dwight P. Robinson & Co., Inc., engineers and constructors, in September, 1919, therefore only fifteen months was required to put the first 60,000-kw. unit into service. This is a record for power-plant construction, considering that the work had to be done under the most adverse conditions, due to the unsettled state of labor and the difficulties in obtaining deliveries of material.

Simplicity of design has been adopted as a fundamental policy consistent with maximum practical reliability. An examination of the inclosed insert will clearly indicate the simple and convenient arrangement of the equipment. All the boiler-room auxiliaries, together with the house turbine, are located in a bay on a level with the turbine-room floor. This arrangement makes it possible to keep this equipment clean and facilitates the convenience of operation.

Past experience has emphasized the necessity of safeguarding the coal supply to large power plants as an insurance to reliable operation. In Colfax station the main coal supply is from the Harwick mine over a privately owned standard-gage railroad, although coal can be received by barge on the river or by car on the Pennsylvania R.R. (see Fig. 8). There is also just north of the station and along the railroad running to the mine a storage area of about 1,000 x 400 ft. which will be spanned by a coal storage and reclaiming bridge crane, providing a normal storage for 150,000 tons. An emergency storage may be obtained amounting to 250,000 tons by piling the coal 40 ft. high.

There are now seven Babcock & Wilcox boilers installed, each having 20,876 sq.ft. of heating surface, fired

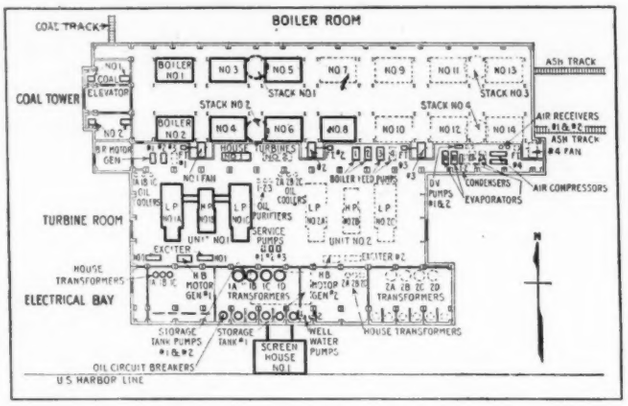


FIG. 9. LAYOUT OF STATION EQUIPMENT

under the low end only, with a 17-retort stoker. Steam pressure of 275 lb. gage and about 180 deg. superheat is used, giving a total temperature of 600 deg. F. The floor below the boiler room is on grade level so that standard steel coal cars can be used to transport the ashes directly from the hoppers under the boilers.

Two self-supporting steel stacks rest on the building structure at the roof line. The tops of these stacks

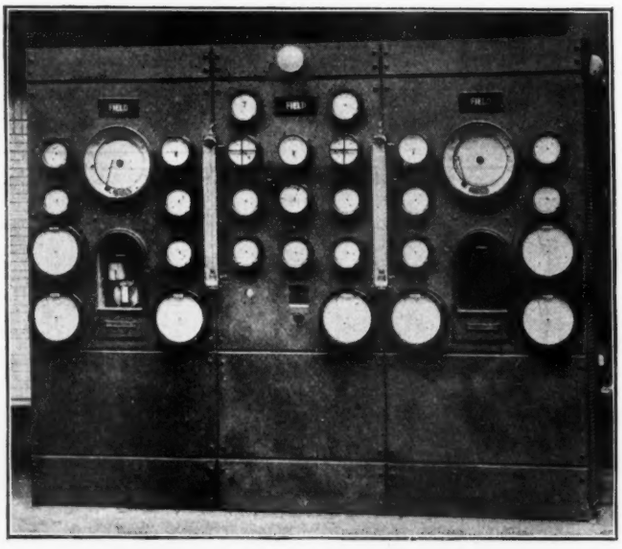


FIG. 10. MAIN TURBINE GAGE BOARD

are 325 ft. above the firing-aisle floor. They are 21 ft. minimum inside diameter and are lined with radial brick.

For many years to come this station will carry the base load of the system and will have a relatively high load factor. A careful study of the all-year perform-

ance indicates that the lowest yearly cost so far as boiler plant is concerned will, for the first few units of the station, be obtained by operating the boilers at between 200 and 225 per cent of rating when the main unit is carrying full load. However, the stokers are extra long and can burn coal at a rate that will allow the boilers to develop 300 per cent at normal rating for peak loads.

Each main turbine-generating unit is of the Westinghouse three-cylinder cross-compound type, designed for a continuous rating of 60,000 kw., with steam at 265 lb. gage pressure at the throttle and 175 deg. of superheat or a total temperature of about 585 deg., expanding to a condenser pressure of 1 in. of mercury. The steam end of the unit has a maximum capacity of 70,000 kw. and with the circulating-air temperatures anticipated in winter months, at which time the yearly peaks occur, the generators will also carry 70,000 kw. at 85 per cent power factor without exceeding the temperature limit of the windings. When operating at its most efficient load (50,000 kw.) the unit will produce one kilowatt-hour on a steam consumption of 10.58 pounds.

The cylinders are so arranged that all steam first passes through a single-flow reaction turbine, operating at 1,800 r.p.m. where, under full load conditions approximately one-half of the total temperature drop occurs. Exhausting from this high-pressure cylinder, the steam is carried by means of overhead pipes to two low-pressure cylinders of semi-double-flow construction, operating at 1,200 r.p.m. Both of these low-pressure cylinders are designed for the same pressure range, and each receives one-half of the total steam. At full load there is approximately the same temperature drop in each of these cylinders as in the high pressure. The pressure distribution is such that when carrying 60,000 kw. the discharge pressure of the first element is approximately 55 lb. gage and the load divides equally among the three elements, each developing 20,000 kw. The general arrangement of the unit is shown in Fig. 2. The axes of the elements are placed on 30-ft. centers, transverse to the length of the turbine room. This spacing affords maximum operating convenience on the main floor and around the condensing equipment in the basement.

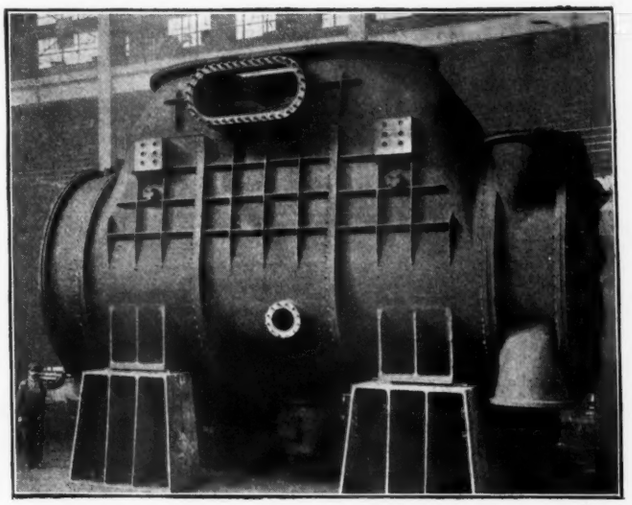


FIG. 13. ONE OF THE CONDENSERS FOR MAIN UNIT

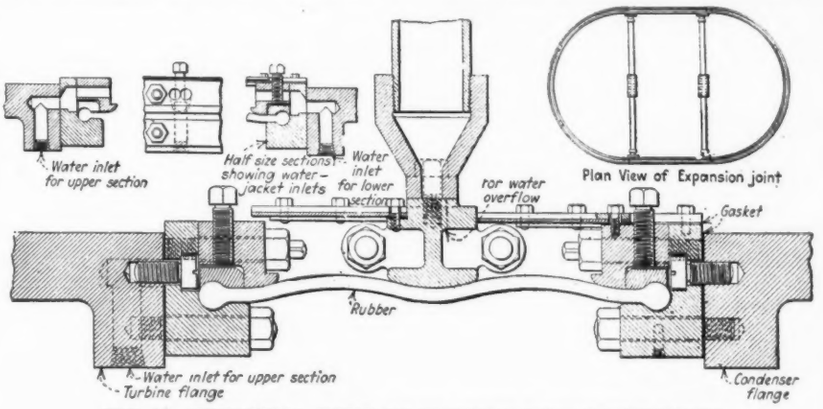


FIG. 11. SECTIONAL VIEW OF TURBINE EXPANSION JOINT

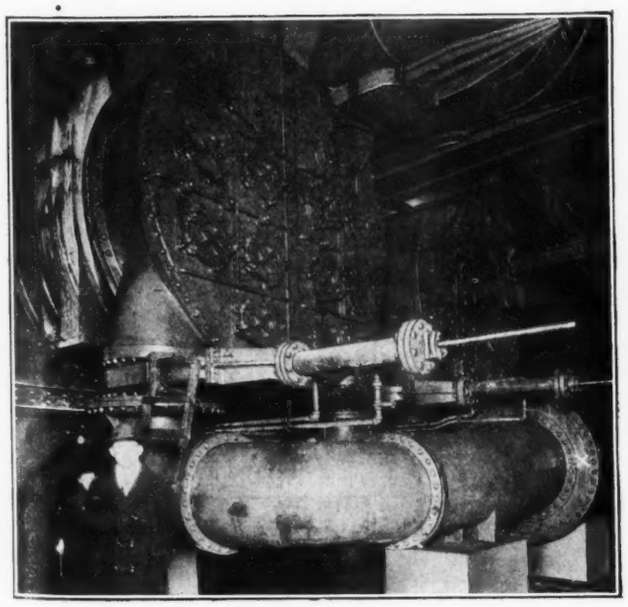


FIG. 12. TWO CONDENSERS INSTALLED

The ultimate station capacity contemplated is 300,000 kw. As the most economical station on the system, it will carry the base load. Therefore, it was desirable to install the largest generating unit developed, with due consideration to economy and reliability of operation.

Through the utilization of the multi-cylinder principle the designer can confine the high-pressure, superheated-steam element to a small rigid structure with speeds and blading suitable to the specific volume of steam under these conditions and can then design the low-pressure cylinders independently, adopting for these low-pressure elements the dimensions and speeds best suited to the steam conditions at the lower pressures.

While the controlling factor in the choice of the multi-cylinder unit is maximum operating reliability, with the efficiency made possible by large units, there is the further very desirable advantage of flexibility in operation. The high-pressure and one low-pressure element may be operated together to carry a load of 30,000 kw. with a steam consumption of 12.6 lb. per kw.-hr.,

or the low-pressure elements may be operated independently, developing a load of 30,000 kw. with a steam consumption of 13.8 pounds.

Normally, all steam to the unit is under the control of the high-pressure element governor, but each low-pressure element is also equipped with a governor and

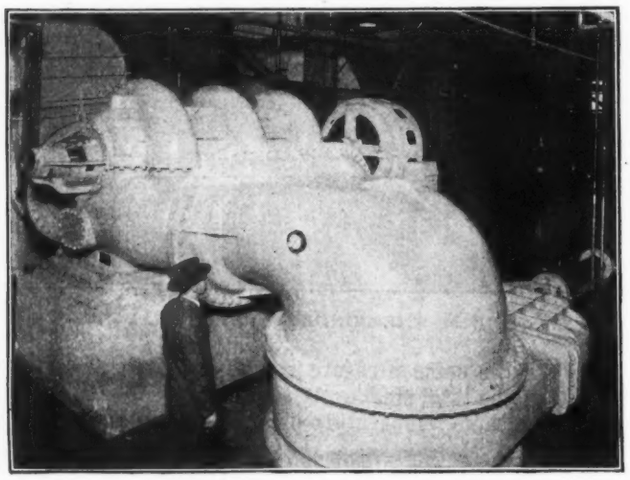


FIG. 14. CONDENSER CIRCULATING-WATER PUMP

a direct steam connection (see Fig. 4). All governors have remote control, and in addition the two low-pressure governors have position indicators on the switchboard.

If, when all three units are operating together, one of the low-pressure elements overspeeds, the governor on that element, normally inactive, will close a butterfly valve in the steam pipe between the high- and the low-pressure cylinders, thus cutting off all steam to this low-pressure element. The excess steam then coming from the high-pressure element is discharged to the atmosphere through a 25-in. relief valve set to open at about 60 lb. This exhaust to atmosphere continues only until the operator adjusts the loads to the new conditions.

If, when the entire unit is operating, the automatic stop valve on the high-pressure element closes owing to overspeed or other causes, thus cutting off all steam from the unit and closing both butterfly valves, the unit then begins to slow down, but as soon as the speed falls approximately three per cent, valves are automatically opened, admitting steam direct from the boilers to the two low-pressure elements, which continue to carry their loads under the control of independent governors. With the high-pressure and one low-pressure element operating together, should the automatic stop on the high function, the low will continue to carry load, taking steam direct from the boilers. Should the governor on the low close the butterfly valve in the steam line between the high and low, the high-pressure element will continue to carry its load, exhausting to atmosphere until the load is either removed or the other low-pressure element is brought into service.

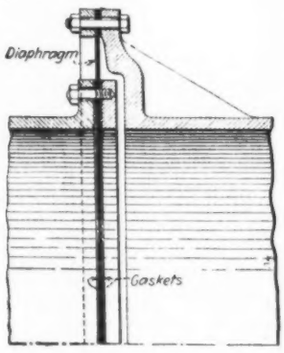


FIG. 15. SECTION THROUGH CONDENSER-PIPING EXPANSION JOINT

The high-pressure spindle is built up of three main pieces of high-grade steel—a central body, or drum, and two shaft ends (Fig. 3). Both end sections are secured to the drum by long spigoted press fits, reinforced at the coupling end by tap bolts and at the governor by shrink links. The low-pressure spindle is built up of a central section consisting of a hollow drum on which are mounted the single-flow reaction blading and the labyrinth packing (Fig. 4). The shaft ends are secured to this drum, and each carries two discs of low-pressure reaction blading.

Fig. 7 shows the low-pressure element with the cover removed. The single-flow stationary reaction blading is carried in a cast-steel section independent of the main casing, as shown in the figure. The central portion of the main cylinder is also of steel, so that in event of operation with steam direct from the boilers, all parts subjected to high temperatures are of steel.

Each unit has its own gage board (Fig. 10) on the main turbine-room floor, on which are mounted all electrical and mechanical instruments needed for its operation and control, including recording steam-flow and condensate meters as well as recording thermometers on the inlet steam, the exhaust steam and the condensate.

As previously stated, the low-pressure elements are of semi-double-flow construction, there being two exhausts from each cylinder with a condenser under each exhaust. In providing the connections to the condensers, it was felt that the ordinary type of copper expansion joint could not be built with sufficient flexibility

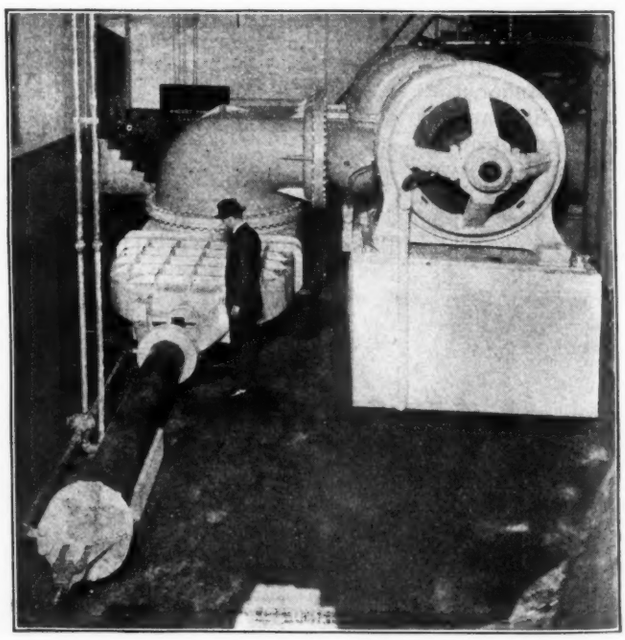


FIG. 16. CONDENSER CIRCULATING-WATER PUMP AND GATE VALVE

n the size required, each exhaust opening being an elliptical section approximately 9 x 14 ft.; therefore, a connection was made by means of an elliptical tube of $\frac{3}{8}$-in. molded rubber two feet in length, the ends suitably formed for clamping to the turbine and condenser flanges (Fig. 11). A further advantage of this rubber joint is that a replacement may be made with-

out dismantling either the turbine or condenser, as a manhole in the turbine exhaust will admit the rubber section and the bolting may be done inside the exhaust chamber. The joints for the first unit were installed in this manner after the turbine and condenser had been set in place. Provision is made for water cooling the joints to keep the operating temperatures within safe limits for the rubber.

Each main generating unit, which has a normal full-load steam consumption of 648,000 lb. per hour, has 100,000 sq.ft. of condensing surface arranged in four shells, containing 25,000 sq.ft. each, there being one for each turbine exhaust. In each condenser there are 5,070 one-inch Muntz metal No. 16 gage tubes, each 18 ft. 10 in. in active length. Each shell is so divided that one-half of it may be isolated for cleaning, thus permitting one-eighth of the total surface to be cleaned at a time without interruption to service. This is considered a particularly desirable feature for inland river locations, where freshets may cause an enormous amount of leaves and other small débris to be present in the water, which, in spite of all reasonable precautions in the way of screens, will find their way into the condensers and are likely to cause interruptions to service unless means are provided for convenient cleaning. Fig. 13 shows a shop view of one of these condenser shells and plainly indicates the arrangement of piping for the divided waterbox construction. The opening on the side near the top is for an equalizing pipe between the two condenser shells serving one low-pressure unit. The pads just below this opening are for the straps which suspend the condenser from the steel framework supporting the turbines. The feet on which the condenser is shown to be supported are for convenience in erection only.

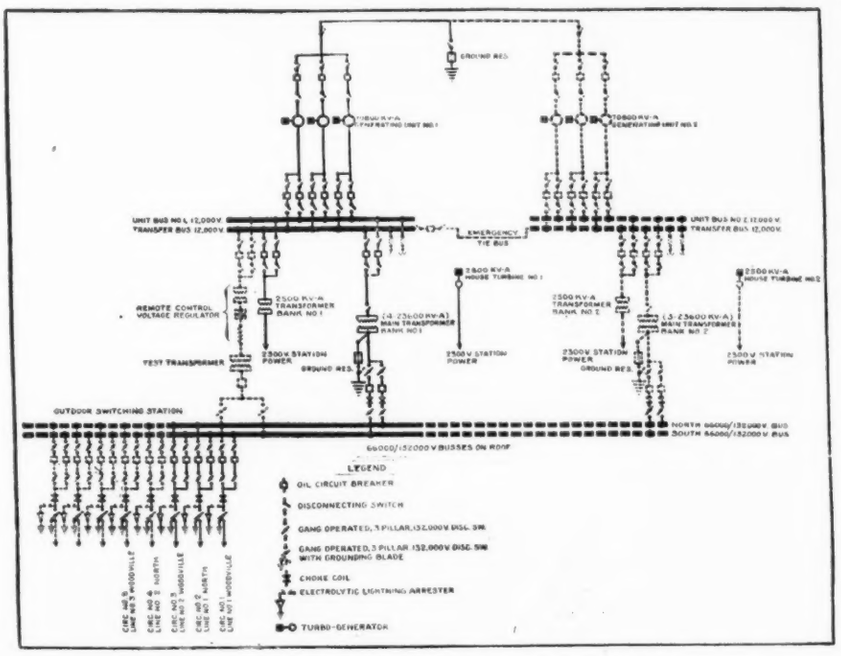

GROUND RES.
UNIT BUS NO 1, 12,000 V. TRANSFER BUS 12,000 V.
EMERGENCY TIE BUS
UNIT BUS NO 2, 12,000 V. TRANSFER BUS 12,000 V.
REMOTE CONTROL VOLTAGE REGULATOR
2500 KV-A TRANSFORMER BANK NO 1
2500 KV-A TRANSFORMER BANK NO 2
2300 KV-A HOUSE TURBINE NO 1
2300 KV-A HOUSE TURBINE NO 2
TEST TRANSFORMER
2300 V STATION POWER
GROUND RES.
OUTDOOR SWITCHING STATION
NORTH 66000/132000 V. BUS
SOUTH 66000/132000 V. BUS
66000/132000 V BUSSES ON ROOF
LEGEND
OIL CIRCUIT BREAKER
DISCONNECTING SWITCH
CHOKE COIL
ELECTROLYTIC LIGHTNING ARRESTER
TURBO-GENERATOR

FIG. 17. ONE-LINE DIAGRAM OF MAIN ELECTRICAL CIRCUITS

For each 100,000 sq.ft. of condenser installation there are provided three motor-driven circulating pumps discharging into a common header so that any pump or combination of pumps may be operated. Each pump has a capacity of 44,000 gal. per min. at a speed of 480 r.p.m., with a suction pipe equivalent to 72-in. diameter and a discharge pipe of 60-in. diameter. Each pump is driven by a 600-hp. 2,200-volt induction motor.

There are installed four motor-driven two-stage 1,000-gal.-per-min. condensate pumps, designed for 150-ft. external head to discharge through the evaporator condenser to the head tank for the barometric feed-water heater. With maximum load on the turbine only two of these pumps are required, the others serving as spares as well as to meet such an emergency as excessive leakage due to burst tubes. A 100-hp. 1,100-r.p.m. 2,200-volt induction motor drives each pump.

Computations involving the varying vacua throughout the year due to the changes in circulating-water temperature indicated a smaller steam consumption for air removal by using motor-driven hydraulic pumps in connection with a house turbine than with air ejectors of the steam-jet type. Three pumps of the Leblanc type, using raw river water, are installed. These pumps obtain their hurling water from two concrete tanks beneath the basement floor and discharge into the same tanks, with sufficient water travel between discharge and suction pipes to insure complete air separation. In order to maintain constant water temperature, a makeup water line is led into these tanks and two motor-driven pumps installed to take care of the overflow. The operation of these removal pumps and the supply of makeup water is regulated by a float control. It is further planned to install a rotating dry-vacuum pump in connection with an air bell for the purpose of measuring the leakage. A saline testing apparatus is regularly used for detecting any leakage of river water into the condensate. The vacua anticipated with condensing water at an inlet temperature of 60 deg., which is higher than the average temperature of the river, are as follows:

Pounds of Steam per Hour	Vacuum in Inches of Mercury
400,000	29.20
500,000	29.12
600,000	29.03
700,000	28.95

Where valves are needed for control or regulation in the circulating-water piping, they are motor-driven, otherwise they are hydraulically operated by means of the boiler feed-water pressure and have either tail rods or other position indicators (see Fig. 16).

Plate or diaphragm expansion joints (Fig. 14) are used between the water piping and the condensers, resulting in a considerable saving of space over the more common forms of expansion joints. One of these joints is shown at *X*, Fig. 12.

Recording venturi meters are installed to measure the condensate, and recording thermometers are used to measure the temperature of both the exhaust steam and the condensate.

A somewhat unusual feature, made possible by the turbine room being parallel to and near the river, is

that there is a separate circulating-water intake and revolving-screen well for each two units. This not only defers the investment in future intake capacity, but will result in the ultimate first cost of the complete plant being less, so far as intake flumes are concerned, than would be the case if an intake common to all units were constructed. For similar reasons of first cost, the discharge flume located in a water-bearing gravel was made wide and shallow, thereby avoiding greater cost due to deeper construction. (See insert of large cross-section for a section through the intake and screen well.)

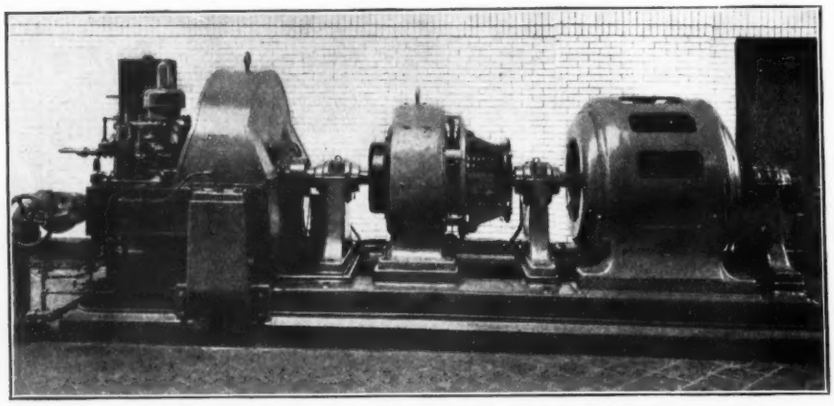


FIG. 18. DUPLEX MOTOR- AND TURBINE-DRIVEN EXCITER

Referring to Fig. 17, a single-line diagram of main-station connections shows the comparatively small number of electrical circuits and switches required for the large plant capacity available. The three 20,000-kw. 12,000-volt 60-cycle 23,600-kva. generating elements of each main unit are operated as one machine, being switched together on a common bus and given field excitation while stationary, then brought up to normal frequency and voltage as one generator by the admission of steam to the high-pressure turbine, and synchronized with the 66,000-volt system through the low-tension circuit breaker of a 70,000-kva. transformer bank. The design of the ultimate station includes no plan to parallel more than the three elements of one generating unit on any common 12,000-volt bus. Thus, to all effects and purposes, the 70,000-kva. generating units may be considered as individual 66,000-volt machines in multiple at this voltage on a high-tension bus, from which transmission circuits are supplied.

The transformer banks and all the high-tension switching equipment and connections in the power station and outdoor switching are insulated for 132,000 volts. Thus, while operation at present is at 66,000 volts, the increase to a higher potential can be made at any time by a simple change in transformer connections when growth in station capacity and system loads requires.

Each unit is electrically isolated from all others. Auxiliaries, excitation, control and other essentials are individual, so that one machine is not involved in internal disturbances affecting another. The absence of 12,000-volt multiple connections limits the short-circuit currents flowing into a low-tension fault to such an extent that standard apparatus and methods of construction can be safely employed in a station where large amounts of generating capacity are physically concentrated.

The ultimate station will house under one roof six power plants of moderate rating. By this combination economies are effected in fixed charges and operating costs, which would be impossible if the plants were physically separated. At the same time, by electrically isolating the individual units, the system is protected against excessive low-tension short-circuit currents. In other stations of large capacity, stresses developed by such short-circuits have at times exceeded the rating of protective devices and resulted in destruction of apparatus and serious interruption of service. At Colfax the arrangement is essentially that of a group of stations in parallel on a high-tension bus where segregation of trouble is secured by the spacing of equipment incidental to high-tension construction. Thus the advantages of concentrated generating capacity are secured with minimum hazard.

For the protection of generator windings the neutral connections are brought out and differential relay protection is provided which, in case of internal short-circuit, automatically disconnects the affected generating element from the 12,000-volt bus and trips the throttle valve, cutting off the steam supply, operates the field circuit breakers and sounds a fire alarm in the turbine room, lighting red signal lamps at the generator and switchboard.

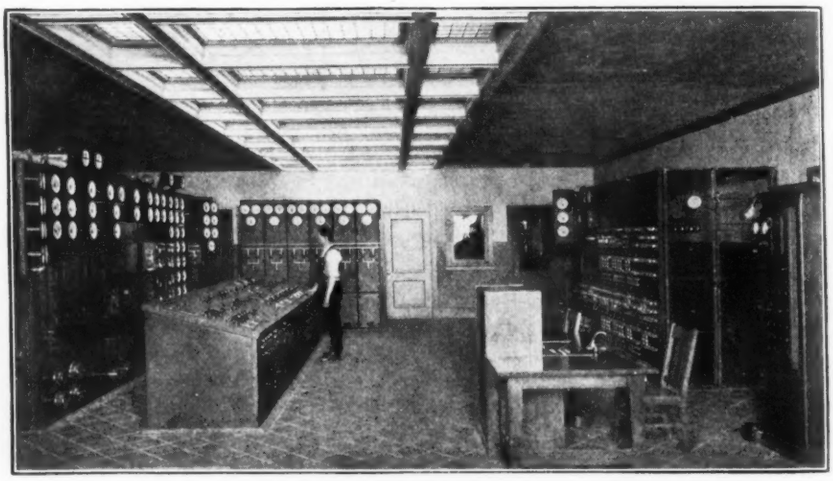


FIG. 19. MAIN SWITCHBOARDS IN CONTROL ROOM

Provisions have been made to extinguish fire in the generators by turning high-pressure steam into the windings. Air-duct dampers can be dropped by remote control from the switchboard room. Both damper trip and fire-extinguisher steam valve can be connected for automatic operation if considered advisable.

A resistance of 7 ohms is installed between the generator neutrals and ground, limiting ground current on the 12,000-volt system to 1,000 amperes in case of complete breakdown. Between the generator neutral resistor and its connection to ground a current transformer is installed. A green signal light and alarm bell in the switchboard room are connected to its secondary. By this arrangement the operator is at once notified of station trouble involving flashovers or breakdowns to ground on any of the 12,000-volt station circuits.

Each generating element is equipped with six resistance coils and six thermocouples for temperature measurement. The resistance coils terminate in switchboard indicators, and the thermocouples are connected to a potentiometer pedestal, also in the control room. A resistance coil in the airduct makes it possible to determine at any time the rise of the windings above temperature of the cooling air entering the machine.

Excitation is supplied by 250-volt 350-kw. duplex motor- and turbine-driven exciter (Fig. 18). Two of these dual drive sets are installed with the first unit, one to be a spare in the ultimate station. Normal operation of the exciter set is by motor drive with a small amount of live steam admitted to the turbine casing. In case of motor failure through low voltage or by trip-out, the resultant decrease in exciter speed

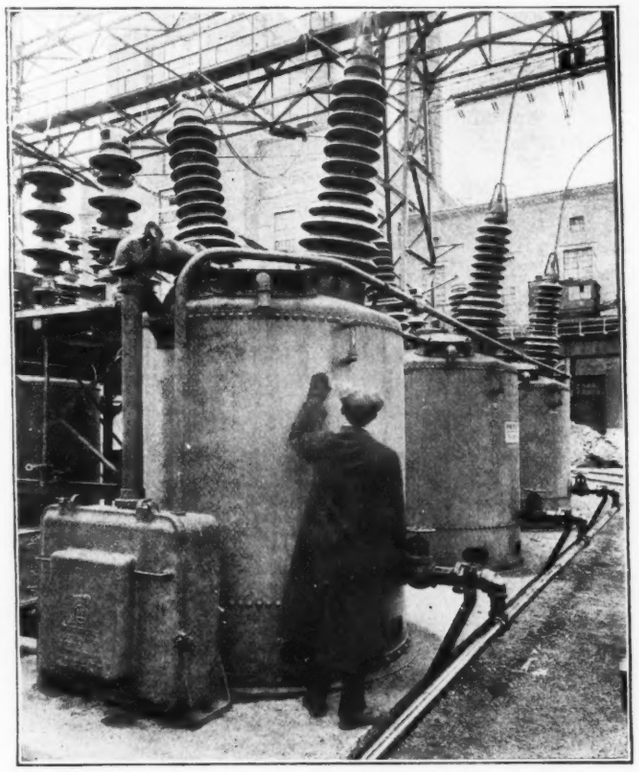


FIG. 20. A 132,000-VOLT 400-AMPERE OIL SWITCH

automatically places the turbine in service as a prime mover, and excitation is maintained without interruption.

Voltage control is secured by either of two methods. The usual vibrating regulator is installed with its contacts working across the rheostat of the shunt-wound exciter. In addition, there is installed what is known as a face-plate regulator, which is an entirely new development and is believed to possess some features of considerable value. This regulator consists of a voltage-control element arranged to adjust through relay switches the position of the motor-operated contact arm of the generator field rheostat.

When the contacts of a vibrating regulator fail to operate properly, the resultant variation in generator

FIG. 21. OIL SWITCH; SIGNAL LIGHTS AT X

voltage from normal may be extreme. With the face-plate regulator this disadvantage is largely eliminated. Absence of multiplicity of contacts and connections increases the reliability of operation, and the regulator can be easily cut in or out of service. The device is inherently simple and is expected to function successfully.

The main benchboard or control desk has been laid out with special attention to operating convenience. The floor under the benchboard has been dropped so that additional headroom is provided. It is possible to stand erect below the desk apron and thus work to advantage on control-equipment connections.

Transit board barriers are provided between adjacent panels, thus confining any burnout of control wiring to a single panel. Multiple-conductor instrument and control cable is used with coloring carefully standardized. Each terminal in any group assigned to a given conduit number is marked with the color of the wire that is to be connected to it. This simplifies the work of originally making switchboard connections and of replacing them in case of routine maintenance work.

Spare conduits are provided for additional circuits in the future in case they are required. Mimic busbars are located on top of the desk apron, a different finish being used for each potential represented.

A safety feature of particular interest is one to prevent operation of the oil circuit breakers while disconnecting switches in series with them are being operated. At each circuit-breaker cell there are located three 15-watt lamps behind red, green and white signal lenses (See *X*, Fig. 21). The red light indicates a closed circuit breaker, the green light that the breaker is open. Directly below the indicating lamps is placed

a small flush-type double-pole push-button switch. The function of this switch is to interrupt the control circuit of the circuit breaker in the compartment, at the same time illuminating the white lens in the signal box above the push button. Thus anyone operating disconnects can see that the breaker is in the open position, and prevent its being closed by mistake.

Insufficient provision for power-station grounding is frequently the cause of serious troubles, involving burning out of control and instrument wiring and prolonged delay in resumption of service. At Colfax, grounding plates and their connections present such a low-resistance path to earth that ground currents will not seek any other routes. Seven ground plates are buried beneath low-water mark on the river side of the station and are connected to a 750,000-circ.mil bare-copper cable ring bus by individual cables of the same size. Inside the building this cable is run exposed, being clamped to walls where cable and joints are at all times subject to inspection. Risers of copper strap make connection to the ring buses on the upper floors of the electrical bay. Individual circuits are run between those ring buses and important pieces of apparatus. Thus a through copper connection of ample size between equipment and ground is provided. Failure of several ground plates by breaking of cable connection to them or even an open circuit in the ring bus itself will not not seriously affect the operation of the grounding system.

Four single-phase 23,600-kva. indoor, oil-insulated water-cooled transformers are installed with the first unit. Three form a delta-star 12,000-volt, 66,000/132,000-volt, 70,800-kva. bank. The fourth stands with all piping connection complete, ready for quick connection electrically in case of failure of one of the operating units. A complete change-over will not require more than 20 minutes.

With the bank again in service the damaged transformer unit which has been disconnected can be moved into the turbine room on permanent rails provided, and there such repairs as are required can be made with the help of the station crane. The transformers which are larger in capacity (23,600 kva.) than any single-phase units yet constructed, are approximately 10 ft. in diameter and 22 ft. high to the top of the 132,000-volt terminals. The efficiency of these transformers at full load is 99 per cent.

After being stepped up to 66,000 volts, the power passes through high-tension switching equipment and then up lead wells at the side of the power station to the 66,000-volt buses on the roof. It will be seen from the cross-section of the station that the high-tension cables leave the building without the use of roof or wall bushings. This feature has increased the reliability of the high-tension circuits. The buses on the roof go directly to an outdoor switching station, Fig. 22, at the end of the plant, where connections are made to the Duquesne Ring.

The author wishes to acknowledge the assistance and co-operation rendered in the preparation of this article by C. S. Cook, General Manager and J. M. Craves, Assistant General Manager of the Duquesne Light Company, and by various members of the organization of Dwight P. Robinson & Company, Incorporated, Engineers and Constructors.

[A second article describing the boiler and auxiliary equipment of the plant will appear in an early issue.—EDITOR.]

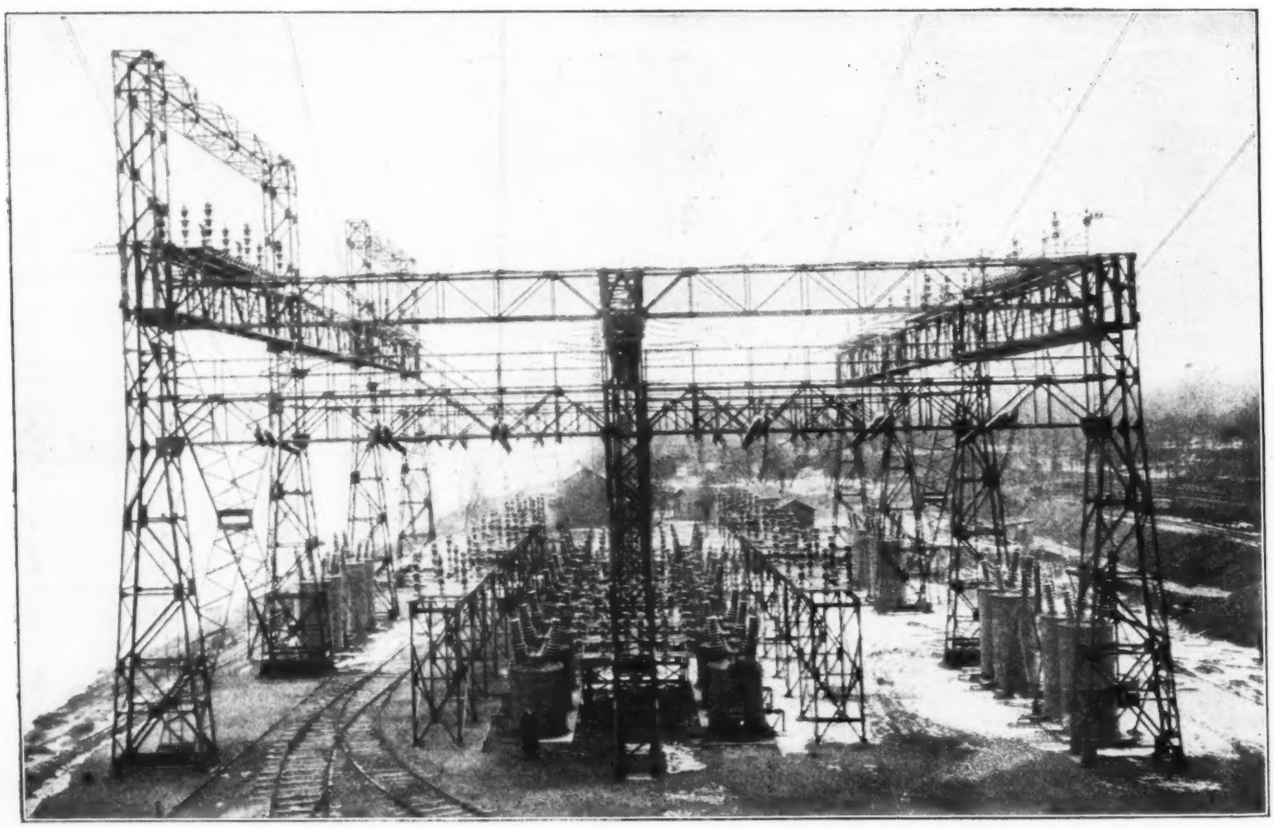


FIG. 22. OUTDOOR SWITCHING STATION FROM WHICH COLFAX POWER IS DISTRIBUTED

Ammonia Compressor-Rod Packing

By W. H. MOTZ
Department of Mechanical Engineering, University of Wisconsin.

WHEN a moving part of a machine or apparatus passes through a wall of the machine or apparatus into a region of higher or lower pressure, a packing is interposed between the stationary and the moving parts in order to reduce to a minimum the leakage of the medium from the higher pressure to the lower pressure. The importance of the proper selection and care of packing may be more thoroughly appreciated when one remembers that approximately 75 per cent of the loss of ammonia in the average compression refrigeration plant may be charged to leakage through the stuffing boxes. It is probably true that there is no other part of the system that requires so much regulation and care and causes so much anxiety as the stuffing box. The difficulties that are generally encountered are largely due to excessive friction or wide ranges of temperature.

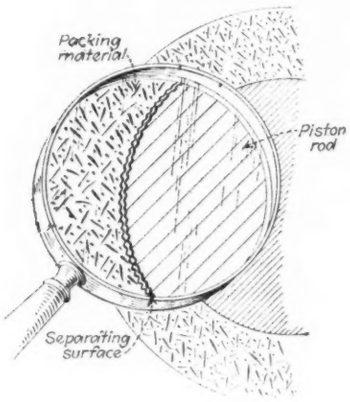


FIG. 1. RUBBING SURFACE OF A PISTON ROD

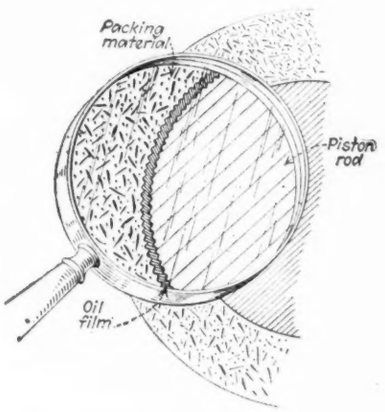


FIG. 2. RUBBING SURFACES SEPARATED BY OIL FILM

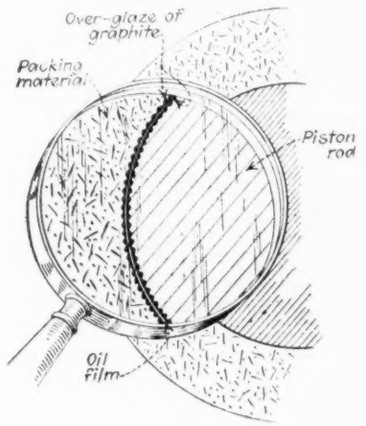


FIG. 3. FRICTION DECREASED BY GRAPHITE

Principle of Packings

When two solid surfaces are held together by any appreciable force, any effort tending to move them relatively to each other—that is, to move or rub one surface over the other—is met by a resisting force acting tangentially to the surface of the two bodies. This resisting of relative motion, commonly termed friction, is due to the interlocking action of the minute depressions and elevations which exist even in the smoothest surfaces, and will vary with the different kinds of materials, finish of surfaces and the speed of the relative motion. The appearance of these irregularities of the surfaces is illustrated by Fig. 1. This figure shows a small section of the separating surfaces between a smooth piston rod and a smooth packing substance, as it would appear under a magnifying glass, It is apparent that if these two surfaces are held in contact by a force, the effort required to slide one over the other is quite large.

Separating Oil Films

Now, if oils or lubricants are interposed between the surfaces, the frictional resistance is materially reduced because the rubbing surfaces are wholly or partly separated by a thin film of the lubricant. Fig. 2 shows how the film of oil separates the packing and rod so that they do not tend to seize. It is obvious that if the rubbing surfaces are made as smooth as possible, the frictional resistance to motion will be lessened and the surfaces will be easier to lubricate. If the depressions are filled with some suitable substance and the surface glazed over smooth, the cause of frictional resistance will be partly overcome. Fig. 3 shows how a packing and rod may be glazed over with graphite and separated entirely by means of an oil film.

The proper functioning of the stuffing box depends upon the presence of this oil film. Owing to the phenomena of capillary attraction and molecular adhesion, the oil film clings to the surface with such tenacity as to support a load as large as 800 lb. per square inch. The exact magnitude of this load intensity is governed by a number of things such as character and type of surface, finish of surfaces, kind of oils, etc. It is apparent that if a film of oil is interposed between the packing and the piston rod, as illustrated by Fig. 4, and that if the packing is subjected to pressure due to forcing the gland into the box, the escape of high-pressure gas from A along the rod to the atmosphere at B is opposed by the resistance of the oil film. This is the fundamental principle of all successful packings. Since the rod is in motion and under vibration, it is necessary to have a considerable length of effective film between C and D in order to prevent the escape of gas of different pressures.

It should be further noted that the employment of frictional substances as packing is inconsistent with sound engineering principles, since a packing consisting solely of frictional materials depends upon the pressure of the substance against the face of the rod to prevent the escape of the medium under pressure.

The effect of varying temperatures of the rod upon the packing should be observed. The compressors in actual operation are subjected to different temperatures due to operating on wet, dry or superheated vapors. The packing should respond to slight changes of dimensions instantly in order to conform to the rod.

This would preserve the oil film and prevent the escape of gas. The packing should then be made of some resilient material that will compensate automatically and instantly for any irregular movement of the rod, or any expansion or contraction due to changes of temperature.

Requirements of a Good Packing

From the foregoing discussion it may be appreciated that to be successful a packing has certain definite requirements that must be fulfilled. It will be observed also that the desired functions of the packing are determined by economic considerations. The following may be considered cardinal properties of good packing:

In the first place, the packing must be gastight. It must reduce to a very small minimum or prevent entirely the escape of the medium under pressure. The packing must remain gastight under the variations of temperature that occur in actual operation. The escape of the refrigerating media increases the cost of operation, since new ammonia must be charged into the system to make up for losses.

In the second place, the frictional resistance of the piston rod, as it reciprocates to and fro through the packing box, should be reduced to a minimum. It is apparent that the energy used to overcome frictional resistance is simply wasted. This energy cannot produce any good and cannot be recovered; therefore, it is desirable to keep this loss as low as possible. Furthermore, excessive friction causes great wear and abrasive action on the surface of the rods, so that they become scored and shouldered. Under this condition larger quantities of lubricating oil are required.

In the third place, the packing must be resilient. It must be sensitive to any change of pressure and temperature or any irregular movement of the rod. It must adjust itself automatically and instantly to the operating condition that exists.

Lastly, the packing should have a fairly long life in order that renewals are necessary only after long intervals of time. It is desirable to replace the packing only after long intervals of time on account of economic considerations. A packing that will function properly for the longest possible time should be selected for the service.

Types of Packings

There are many forms and types of packing that may be purchased in the market. Each type will give more or less efficient service in its respective field of application. The successful use of any packing depends upon the intelligent selection and adoption to the particular service in any case.

Packings may be classified into two principal types, fibrous and metallic. Metallic type packings may be divided into two classes, non-elastic and elastic. Fibrous packing may apply to all substances made of fibers of plants, such as cotton, hemp, jute, flax or any combination of these materials twisted, braided, woven or mixed with rubber compounds and made into packing.

Non-elastic metallic packings are nothing more nor less than bushings made of soft metallic alloys. These packings fit the rods snugly; therefore the rods must be true and in line for efficient operation. They are held against the rod by the application of pressure produced by screwing up the gland.

Elastic metallic packings are usually composed of rings of metal, each cut into two or more segments. These rings are held against the rod by helical or flat springs around their circumferences and are placed in pairs into compartments of a split retaining case that is inserted into the stuffing box. The springs allow the segments to move laterally, maintaining in this manner a tight joint upon the surface of the rod and on the face of the partition in the retaining casing at the same time.

The Merits of Packings

The metallic-type packings have been used generally in the past for service on high-duty apparatus. This type has been found to be dependable and therefore successful in this field. No doubt the local conditions have had an important bearing on the efficient operation of this packing. Since this type depends on the establishment and maintenance of an oil film between the

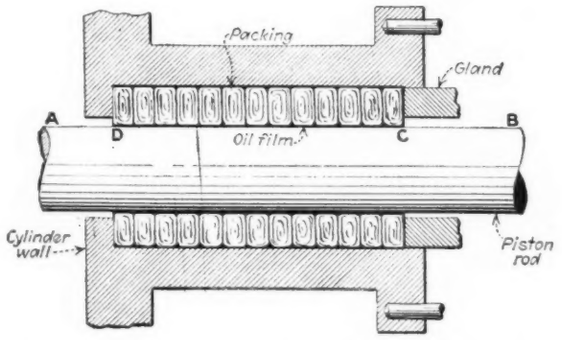


FIG. 1. PACKING SEALED WITH AN OIL FILM

packing substance and the rod, it must be well supplied with lubricating oil at all times. Also, it is necessary that the rod be true and well aligned. Therefore, this type has been in a more or less advantageous position to effect efficient service.

The fibrous or soft packings have been applied usually on the low-duty apparatus, such as pumps, valve stems, plungers, shafts and piston rods. The field of application of fibrous packing is relatively large, since there is much more low-duty equipment than high-duty apparatus. This type of packing gives desirable results on rods that are not true. A large amount of efficient service has been obtained from this kind of packing in the past, and the same performance may be expected in the future.

On the whole, it may be said that the fibrous types are being replaced somewhat by the metallic types. This is probably due to the anti-friction properties and longer life of the metallic types.

The metallic type may be considered somewhat more gastight than the fibrous type. The gastightness of the former depends on the oil seal, while the latter depends more or less on the physical contact of the rod and packing material. The metallic type has less friction on the rod, since the greater portion of the packing that is against the rod is constructed of anti-friction or bearing metals. Also, this type depends on the oil seal to prevent the escape of the medium under pressure rather than the forcing of the packing against the rod to stop the leakage of the gas, which results in reduced friction. The soft packing is comprised of such material as rubber, cotton fabric, asbestos, etc., which have large coefficients of friction. In any case too much packing should not be placed in the box,

so that when the gland is started fair, excessive friction is not produced by the extreme pressure created by the gland. The gland nuts should be only hand-tight.

With changes of temperature within the cylinder, the metallic type is more or less automatic or self-adjusting in compensating for the expansion and contraction of the materials. On the other hand, the fibrous packing will expand and contract with changes of temperatures so that leakage will occur if the glands are not readjusted. However, the metallic types (exclusive of the elastic type) require a much more delicate and careful adjustment of the gland than the fibrous types do, in order to secure the proper pressure on the packing.

Generally speaking, it may be said that the metallic packings are more durable than the fibrous kind. This is due to the fact that the fibrous packings are composed of materials that tend to disintegrate under the actions of friction and heat. In the ultimate analysis it will be noted that the operating engineer is responsible to a large extent for the successful or poor performance of the packing materials, since it is his duty to give to this important part of the compressor its proper care, adjustment and lubrication.

Determining the Safe Working Boiler Pressure

By J. P. Morrison

The highbrow may grow higher as the mathematician again sharpens his pencil, in his endeavor to determine the safe working pressure for a boiler, while from the sawmill country comes a boiler inspector's account of a shorter, more direct and conclusive method, even if the results were not quite all that could be desired.

It appears that this inspector was called to give advice as to repairs needed to place a boiler in serviceable condition. Upon reaching the plant he encountered a horizontal-tubular boiler of butt-strap design, 72 in. in diameter, 18 ft. long, containing seventy 4-in. tubes, and intended for operation at 125 lb. pressure—in fact, just such a boiler as would gladden the heart of an engineer who was operating a comparatively small power plant.

The boiler had been in operation but a short time when the increasing load made an increase in pressure appear desirable, so the operating engineer sought advice of the engineer in charge of a somewhat larger neighboring mill, for whose knowledge and experience our engineer had the greatest respect. The neighboring engineer, no doubt feeling the necessity of maintaining his local prestige as an engineering oracle, but with the best intentions, offered the use of his hand pump, and advised applying a hydrostatic test pressure, which was to be raised until the shell plates were observed to *sweat*, when the pressure registered by the gage was to be noted and recorded as the safe working pressure, or popping pressure; that is, the pressure to which the safety valves could be adjusted to operate.

Our engineer had no idea such an important question could be disposed of so easily, but connected up the pump and proceeded as directed. The atmospheric conditions or the temperature of the water in the boiler, singly or jointly, may have contributed to the results, but the boiler had not started to *sweat* when the needle on the pressure gage had reached the limit of travel and refused to go farther. While the engineer was satisfied with the pressure he was then sure he could safely carry, he resembled the man with a million dollars who "could not spend it all, but wanted more," so continued to pump, and the inspector is convinced he would be pumping yet had the seams not developed leakage.

The inspector's report indicates the heads, tubes and braces had changed position but slightly, although the shell plates resembled so many barrels. The head seams and girth seams, although distorted and open, had resisted distension, so the shell plates at the seams retained the original diameter, while midway between the girth seams the increase in diameter approximated three inches. This test, if properly conducted to establish the strengthening value of girth-seam construction, would have been a success, but was a failure so far as determining the safe working pressure of the boiler was concerned.

A New Boiler Is Ordered

The inspector's verdict was, "repairs impossible," so the sawmill operator telephoned the boiler builder, "Send me a new one just like the last, but set the safety valve to the safe working pressure and lock it." At this method of determining the safe working pressure the intelligent engineer may well be amused and may decide to stick to his pencil, scratch pad and formula, assisted by a careful hammer test. Ignorance of the purposes, advantages and disadvantages of the hydrostatic test is not confined to backwoods engineers, for while the theory of a boiler *sweating* when the hydrostatic pressure applied corresponds to the safe working pressure may not have gotten very far from the "tall-uncut," it is not at all unusual for the owner of a plant in an industrial district to base a request for an increase in pressure upon the fact that his boiler had been subjected to a certain cold-water pressure, to which, according to his idea, the working pressure should bear a fixed ratio. Apparently, he fails to understand that the safe working pressure is first determined by considering every detail of the material, design and construction of the boiler, and then the hydrostatic test pressure fixed at an amount 33 or 50 per cent in excess of the maximum steam pressure to be carried.

By way of example: Consider a 72-in. diameter boiler of A. S. M. E. Code construction, which the inspector determines is of suitable design, material and workmanship to justify a safety-valve pressure of 150 lb. The boiler is to be subjected to a hydrostatic test, as is usual with new construction, so the inspector applies 225 lb. water pressure, that being 50 per cent greater than the maximum steam pressure permissible, but should the water pressure reach 300 lb., he would not increase the steam allowance that had already been determined.

So much has been written on this subject, it would appear that confusion could no longer exist, but a handbook on engineering which came to the writer's attention recently and was represented as being of 1915 copyright, contains the question, "How is the hydrostatic test applied?" and the answer: "By closing the pipe openings in boiler and filling boiler with water to safety valve. The safety valve is now blocked, and enough more water is pumped into the boiler until the steam gage registers the desired amount of pressure. Boilers so tested *are allowed to carry two-thirds of the test pressure.*" So we may well inquire, "If gold rusts, what will iron do?"

Adjustment of Crosshead-Pin Bearings and Main Bearings of Marine Diesel Engines

By LOUIS R. FORD

THE construction of the crosshead in most large marine Diesel engines follows conventional marine steam-engine practice, the two crosshead pins projecting from the forged crosshead block and the upper end of the connecting rod being forked to support two sets of bearing boxes. In most existing engines the space around the crosshead is restricted to such an extent that it is not possible to use chain falls or other lifting gear when adjusting; but this does not act to handicap the work, for in most cases the crosshead-pin boxes are small enough to be readily handled by hand.

The crosshead being worked on is jacked down to its lowest position and leads inserted to measure the pin clearance. The leads, instead of being laid in the lower box, as is the case with the crankpin bearing, are laid across the top of the crosshead pin, the reason for this being that the weight of all the moving parts above the connecting rod is supported by the lower boxes, and when the engine is stopped the clearance is all on top of the pin. After the leads are taken and before any adjustments are made, the babbitt lining in the lower box should be examined for signs of uneven wear, wiping, cutting, etc. This can be done by hanging the crosshead slipper on the stop bar through the guides and jacking the crankshaft around until the connecting

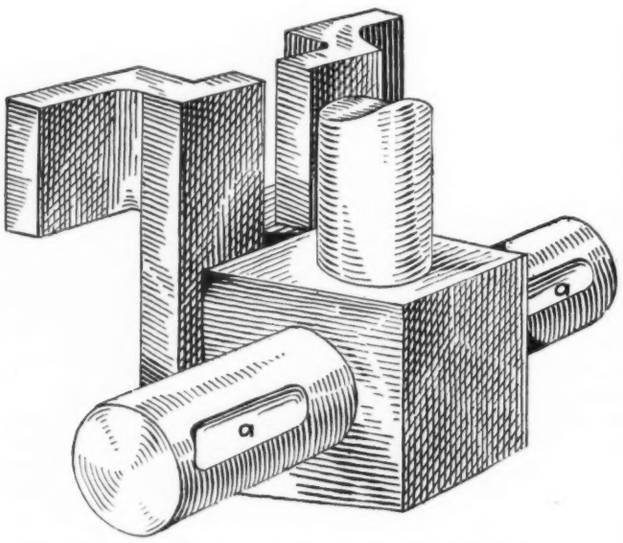


FIG. 1. ELIMINATING SHOULDERS ON CROSSHEAD PIN

rod moves away from the crosshead pins far enough to expose the bearing surface of the lower boxes.

If the engine has been in service long enough to warrant expectation of appreciable wear of crosshead pins, the pins should be examined for shoulders. Unlike the crankpins and main bearings, the crosshead pins do not have a rotative motion within the bearings. As the lower end of the connecting rod rotates in a circle the radius of which is the crank radius, the upper end oscillates so that the boxes turn on the crosshead pins in one direction during half a revolution of the crank and in the opposite direction during the remainder of the revolution. This rubbing of the boxes back and forth on the pins will sometimes cause a shoulder to wear on each side of the pins, and it is necessary to file this shoulder down before the bearing can be properly adjusted. A very good way to prevent the occurrence of these shoulders is to cut away the pin so that there is a flat spot on each side, as shown at *a* in Fig. 1. This allows the bearing surface of the boxes to override the edge of the cut each time, and no shoulders can form.

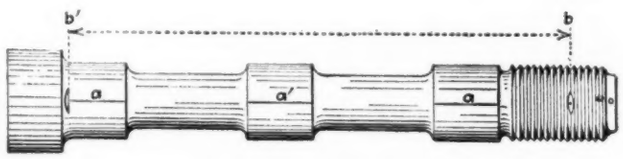


FIG. 2. DETECTING STRETCH OF BOLTS

This does not reduce the effective bearing surface of the pins, because this effective surface is only on the top and bottom portions of the pin.

In case the condition of a crosshead pin bearing is found to be such that scraping and refitting are necessary, great care must be exercised to see that both pairs of boxes on the same crosshead are set up the same amount, otherwise the connecting rod will be "cocked." This, of course, applies only to the forked-end type of connecting rod. The alignment of the rod may be tested after adjusting the bearings by disconnecting the crankpin end of the rod and allowing it to swing free while the crosshead is supported by the bar through the guides. By measuring the clearance between the side of the connecting-rod foot and the inner face of the crank web on both sides, it will be seen whether the connecting rod points forward or aft. This is known as "pointing" the connecting rod. When this pointing shows that the rod is out of line, the engineer should investigate further to make sure that the misalignment is not caused by the whole rod, crosshead and piston assembly being out of line. Another source of error in making this test is the end play of the crankshaft. Be sure that the shaft is in the middle of its fore-and-aft travel.

In the trunk-piston type of engine it is quite common practice to fit the connecting rod in the piston by passing the piston pin through a bronze-bushed hole in the end of the rod without providing any means of adjustment. When wear sufficient to require readjustment occurs in this kind of an engine, the adjustment involves fitting a new piston pin or a new bushing.

Not only does sledging up stretch the bearing bolts, but it also twists them. A number of failures of crosshead bolts have occurred, and in order to guard against these failures the following procedure, which has been used quite extensively, is recommended for both crosshead and crankpin bolts.

On the new bolt scribe a line, as shown at *aa'* in Fig. 2. File a flat spot on one of the threads, as at *b*, and make prick punch marks at *b* and *b'*. Measure the distance *bb'* accurately and record it. Whenever the bear-

ings are adjusted or when the bolts are removed for any other purpose, they should be examined. Any permanent twist will be shown by laying a straight-edge on the line *aa'*, and any stretch will be shown by an increase in the length of the line *bb'*. Any bolt in which the stretch amounts to 2 per cent of its length or the twist exceeds $\frac{1}{8}$ in. in a diameter of 1 in., should be discarded and a new bolt installed.

For measuring the clearance in main bearings, the procedure is practically the same as in the case of the

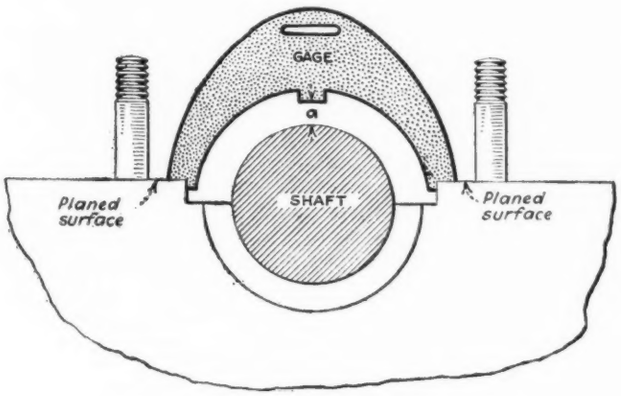


FIG. 3. GAGING MAIN BEARING WEAR

crank and crosshead bearings. The top cap of the bearing being worked on is lifted off with chain falls and the leads are laid on top of the shaft, after which the cap is replaced and the nuts sledged up to their former position. Then the cap is again removed and the thickness of the leads measured. These bearings usually wear very little and, after the initial wearing in, should not need adjustment very frequently. They should be carefully watched, however, for if much wear does occur, the alignment of the shaft is disturbed. The leads taken indicate the total amount of clearance between the shaft and the babbitt lining of the top half of the bearing, but they give no information as to how much of this clearance is due to wear of the bottom half and how much to wear of the top half. A very good indication of the shaft being out of line, owing to having worn down in its bearings, is a variation in the distance between the crank webs when on top center and when on bottom center. If the shaft is in perfect alignment, the distance between the inner faces of the crank webs will be exactly the same for all positions of the crank, but if one end of the shaft is lower than the other, the crank webs will spring open when the crankpin is on bottom center.

A ready means of measuring the amount the shaft is below its original position is available if the builder has supplied a bridge gage. Some builders at the time the engine leaves the shop stamp the bridge-gage readings on the engine bed near each bearing and supply the engine buyer with the gage with which these measurements were made. This gage is similar to the one shown in Fig. 3, and the measurements recorded are the distances between the top of the shaft at each bearing and the point *a* on the gage. Any subsequent measurements with the same gage, when compared with the original measurements, show how much the shaft is down at each bearing. If such a gage is not provided with the engine, the engineer should make some sort of gage and measure the shaft at each bearing at the earliest opportunity, preferably when the engine is being installed, and these measurements should be recorded for future use.

The lower half of the main bearing box is usually made in the form of a circular shell, or cylinder, which fits into a circular recess planed or bored in the crossmember of the engine bed. To remove this box for examination or fitting, a jack is placed under the shaft as close to the bearing as possible and screwed up until the weight of the shaft is supported by the jack. The bearing shell, or box, can then be rolled out of its seating by rotating it around the shaft. If this lower box is to be raised, in order to compensate for wear, shims of the necessory thickness are inserted between the bearing shell and its seating and rolled into place with the shell. On some makes of engine a wedge adjustment is provided under each main bearing, but the value of this is doubtful. The ease with which the crankshaft may be thrown out of alignment with this device may cause it to do more harm than good. It will usually be found that the main bearing nearest to the flywheel will be worn down more rapidly than the others. This is due to the extra weight of the flywheel which has to be carried by this bearing. For this reason the bearing must be watched very carefully.

With very large engines short mandrels are usually supplied, one each of the same size as the main bearings, crankpins and crosshead pins. When refitting a box that has been badly worn, overheated, or rebabbitted, the work can be done at any convenient place in the engine room, using the mandrel for spotting in. This makes it unnecessary to do the spotting on the bearings itself and does away with a lot of laborious handling of the box within the restricted space in the housing.

Determining Carbon Percentage in Lubricating Oil

Only the largest users of lubricating oil are in a position to make a laboratory test to determine the amount of free carbon in an oil that has been used. Often oil used in internal combustion engines will show a decided black color due to the presence of carbon or asphaltum. Frequently, this oil is deemed to be unsuited for further lubricating purposes and is thrown away. Even a very small percentage of carbon will discolor an oil, although the serviceableness of the oil is not injured to any great extent.

To obtain a rough approximation of the amount of carbon present, a drop of the oil can be placed on a filter cloth; if this is not handy, a blotter will do. The clean oil will be drawn away by the capillary action of the paper fibers. The carbon is left and its amount in comparison with the size of the original drop of oil will give a rough idea of the percentage of carbon present. This should not exceed 1 per cent. The term carbon is a misnomer. The deposits in engine cylinders usually called carbon are in most cases made up of an asphaltic gum or tar mixed with grit, dirt, metallic rust and other foreign substances. It is the presence of these substances that constitutes the major objection to the use of a lubricating oil carrying much so-called carbon.

"Self-induction" in an electric circuit is a phenomenon which tends to resist any change of current in that circuit, either in amount or direction. It is due to corresponding changes in the energy stored in the magnetic field set up by the current, and has been described as "magnetic inertia" for that reason.

Considerations in Motor Applications

Proper Relation of Power Factor and Efficiency to Rated Load—Defining the Rated Load of a Motor—Motor Loads May Be Grouped in Three Classes—Heating Effects of Variable Loads—Heat Overload Capacity a Necessity

By PHILIP CHAPIN JONES
Electrical Engineer, B. F. Goodrich Tire and Rubber Company of California

TO DEFINE anything is to express its limits in all directions. To define motor rating is thus to express the relation of the rated output of the motor to all its other variable characteristics. These are temperature rise, efficiency, power factor and speed. Torque need not be included if speed and output are, for it may be derived from them. Of these four characteristics speed may soon be dismissed, as for a standard motor it is preferable to keep it as high and unchangeable as conditions will permit. Temperature rise may also be summarily disposed of, as it is merely a negative requirement specifying that at full-load operation the temperature of the motor shall not rise more than a previously agreed upon number of degrees above that of the surrounding air.

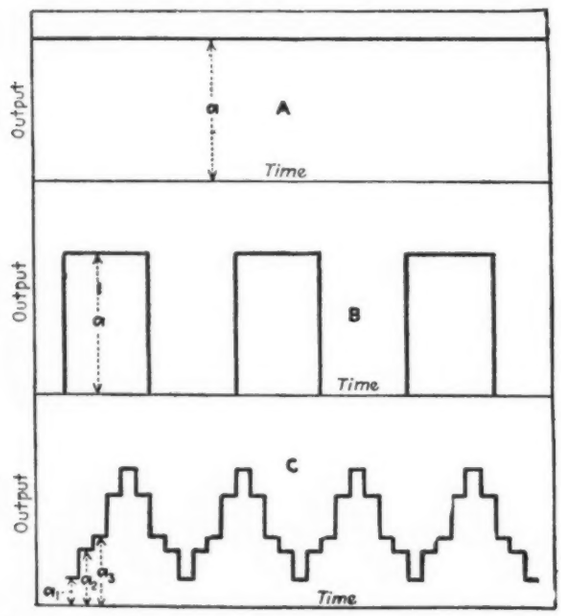


FIG. 1. CURVES SHOWING THREE CLASSES OF LOADS ON MOTORS

The proper relation of power factor and efficiency to rated load will be best determined by considering the case of a motor operating under a load with 100 per cent load factor, such possibly as driving a centrifugal pump. This 100 per cent load-factor load is the simplest type, and as it is one about which all elements may readily be determined, it is obviously the one on which to base a logical definition of rating. It is desirable that over a period of time the average power factor and efficiency should be as high as possible, and as a motor operating at full-rated load has been assumed, the point of maximum efficiency and power factor should thus occur at full load.

Thus the rated load of a motor should be defined as that load at which the rate of change of speed with change in load is as small as practical, and at which the maximum point of the efficiency curve occurs, it being further understood that at this rated output the temperature of the motor shall not rise more than a stated number of degrees above the room temperature. The power factor may well be omitted from the definition because the shape of the power-factor curve is a dependent rather than an independent factor. The curve may be moved bodily up or down by changes in the magnetic circuit which practical considerations generally determine, but its form is inherently fixed and not independent of the efficiency.

Classification of Motor Loads

All the loads a motor may be called upon to drive may be roughly grouped into three classes. These are illustrated in the three curves shown in Fig. 1. The curves shown are merely formalized representatives of their respective classes. 1*A* is of the simplest form, being continuous, with a 100 per cent load factor. It is on such a load as this that the foregoing definition of rating was based. If the peak load of 1*A* is 10, it is obvious that a motor of rating 10 should be selected for this load. The criterion in this case, as in all others, is to select a motor which over its entire time of operation will give a maximum average efficiency and power factor without having its temperature rise exceed the stated limit.

In case 1*B*, assuming the same peak of 10 and the same criterion for selecting a motor, it is also obvious that a motor of rating 10 should be selected. Here, however, owing to the fact that the motor is operating only part of the time, the temperature rise will never reach the allowable limit and it would, therefore, be possible to economize and use a motor with less material. This gives rise to the short-time motor ratings. The short-time rated motor selected would still be essentially of rating 10 as its maximum efficiency would occur at that point, but it would have less heat capacity.

Motors Applied According to Rating

If instead of using a short-time rated motor on the class 1*B* load a smaller continuous rated motor had been used—say of rating 7.5—the same end would not have been attained, for during its period of operation it would be operating beyond the peak of its efficiency curve and thus at a lower value. From the heating viewpoint alone the 7.5 rated motor would have served very satisfactorily, but there would be a loss in efficiency.

Turning now to the load depicted in Fig. 1*C*, it will be observed that an entirely different state of affairs exists. Here the peak is still 10, but the load varies continually from some minimum value up to 10. In

order to be able to say what rating motor would be best suited to this load, it will be necessary to advert for a few moments to observe the general shapes of efficiency and power-factor curves.

A typical efficiency curve is shown in Fig. 2. This curve is actual test data from a 5-hp. 1,200-r.p.m. motor, but it is typical and would apply equally well, except in its absolute values, for any motor. The efficiency of a motor is represented by the expression,

$$Efficiency = \frac{W}{W + L}$$

where W stands for output and L for losses. A plot of this expression, with L constant, will give a curve following that in Fig. 2 very closely except that it will not fall off beyond 100 per cent load, but will approach some maximum value where it will remain practically constant. The falling off beyond 100 per cent load is due to L not being constant—one part of it varying as the square of the current.

From an inspection of this efficiency curve it is evident that if points over the entire range are to be taken and it is desired to have them average up, the average load should occur at or slightly above full load. The fact that the curve does not fall off so rapidly beyond 100 per cent load as it does below, makes an average load a little above 100 per cent give the best average efficiency. An inspection of the power-factor curve, also shown in Fig. 2, makes it even more evident that to obtain the highest average power factor the average load should be somewhat above 100 per cent load.

Heating Effect of Load on Motor

Returning to the problem of fitting a motor to a load of type 1*C*, in the light of the foregoing it is evident that the average load of the curve should be at or slightly above the rated load of the motor. To simplify matters, assume that a motor is selected whose rated load equals the average load. The next question that must be considered is what will be the heating effect on the motor of this load. The average of any load curve is the summation of all the ordinates divided by the number of ordinates or, referring to Fig. 1,

$$Average\ load = \frac{a_1 + a_2 + \text{----} a_n}{n}$$

where n is the number of values taken.

The heating effect, however, as it is dependent mainly on the square of the current, may be roughly represented by the expression,

$$Heating\ effect = \frac{a_1^2 + a_2^2 + \text{---} a^2_n}{n} \qquad (1)$$

The heating effect of the average load would thus be represented by the expression,

$$\left(\frac{a_1 + a_2 + \text{----} a_n}{n}\right)^2 \qquad (2)$$

Unfortunately, for electrical machinery and particularly so for the maximum-rated motor, expression (1) is always greater than expression (2). The ratio of expression (1) to expression (2) may be anything from 1 to infinity, but for ordinary loads is not more than 1.25. For the curve in 1*C* it is about 1.19. The result of this fact is that if a motor of rated load equal to the average load is applied in such a case as 1*C*, the temperature of the motor will rise above its rated temperature. Herein lies one of the great weaknesses of the 50-deg., or maximum-rated, motor which cannot be subjected to a temperature greater than that following continuous operation at full load. For such a load as 1*C* there is, thus, a necessity for a heat overload capacity and this necessity is not accidental or due to practical difficulties, but is inherent in the operating characteristics of the motor. It cannot be escaped. If the point of maximum efficiency were moved back to about 80 per cent of full load so as to allow the motor to operate at 80 per cent rating and still at the point of maximum efficiency, it would not improve conditions, for then if the motor were operated on a constant load equal to its rating, it would be operating inefficiently.

In addition to this heat overload capacity which is inherently required by the 1*C* type of load, all three types require a heat overload capacity to allow for accidental operating difficulties. Dust and dirt will accumulate in the windings, thus blocking the air

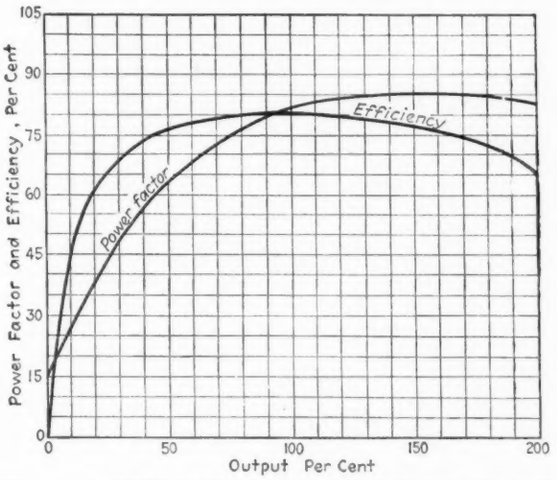


FIG. 2. POWER-FACTOR AND EFFICIENCY CURVES

spaces and reducing ventilation. Machine or shafting bearings may temporarily be binding. New belts may be put on and stretched too tight. Mechanical difficulties may and invariably do arise in the life of a motor which will temporarily throw extra load upon it and thus cause overheating. If it has not a heat overload capacity, it will burn out.

This practical necessity for a heat overload capacity is becoming daily more necessary as motors are being more carefully fitted to their load. Motors used to be selected largely by guesswork and thus were generally too large. It was merely a case of playing safe. With each year now, however, more attention is being paid to the question of motor sizes, as the evils of poor power factor are being realized by the industrial plant and often penalized by the central station. Thus motors are more and more being operated near their rated load, and at this point the necessity for a heat overload capacity to allow for temporary and accidental overloads becomes more pressing.

There is yet another reason inherent in the motor characteristics demanding a heat overload capacity. Referring to Fig. 2, it is obvious that much less is lost in the way of poor efficiency and power factor by operating a motor a given distance over full load than by operating it the same amount underload. This is especially true in the case of power factor which always rises above its rated-load power factor and does not

start bending down till a considerable overload is attained. Realizing these facts, what is the watchful guardian of plant efficiency to do when, as in practically all cases, the exact average is not known or varies within certain limits? He will prefer to apply a motor that is perhaps slightly too small rather than one that is too large. This may safely be done and a substantial gain made in power factor if the motors used have a heat overload capacity, but not if they do not.

Considering the general question of safe temperature for insulating materials, we find in the A. I. E. E. Standardization Rules, in about the middle of paragraph 300, this sentence: "When safe limits are exceeded, deterioration is rapid." The first sentence in paragraph 301 reads thus: "The result of operating at temperatures in excess of the safe limit is to shorten the life of the insulating material." In the first sentence of paragraph 302 are the words: "There does not appear to be any advantage in operating at lower temperatures than the safe limit, so far as the life of the insulation is concerned." These safe temperature limits are given in Table III of paragraph 376 for three classes of materials.

Safe Temperature Limits

There are two implications in these quoted sentences, both of which seem doubtful. The first is that the effect of temperature on insulating materials is zero up to a certain point and very great from that point on; that, in other words, we have a definite break at the safe limit temperature. As a matter of fact this is a most unusual thing in nature. There are cases such as the boiling point of liquids, for instance, where a definite change at a fixed point occurs, but the suddenness of this change is more apparent than real. The liquid is evaporating at all temperatures with increasing rapidity as the temperature rises. The process is quite pronounced even at temperatures considerably below the boiling point.

The second implication is that this safe temperature limit has been definitely fixed. This seems particularly doubtful in view of the recent proposed change in these safe temperature limits by the Power Club.

It is obvious that if the deterioration of insulation does not begin suddenly, as indicated by the A. I. E. E. Rules, but is going on to some extent even below the safe limit, in good motor design a reasonable distance should be kept from this safe limit to lengthen the life of the insulation. If, moreover, this safe-limit point is not definitely located, there is still more reason for allowing ample factor of safety in the motor design. Accepting the A. I. E. E. statements as indisputable—that at a certain temperature, which has been definitely located, the insulation begins to deteriorate very rapidly—does not the very sharpness of the break demand a substantial safety factor? It is like the ultimate strength of a metal. A designer would not think of laying out a bridge using a factor of safety of one, as the supporters of the 50-deg. motor would have motors designed.

Maximum-Rated Motor

A maximum-rated motor is not suitable for general purposes, since it is one with no heat overload capacity, while in the foregoing four reasons were shown why a heat overload capacity is necessary. These are: (1) Wherever in design maximum conditions are dealt with, a factor of safety should be used to reach working conditions. This is as true in regard to maximum temperature of insulating materials as it is with the ultimate strength of metals. (2) All motors should have an overload capacity due to temporary accidental overloads which may be thrown upon them. (3) Owing to the fact that the heating effect of a varying load is greater than that of a constant load equal to the average value of the varying load, a heat overload capacity is necessary if motors are to operate under most efficient conditions. (4) Owing to the shape of the efficiency curve, which falls off more slowly beyond full load than below it, and of the power-factor curve, which rises to a maximum at some point beyond full load, better operating conditions obtain on varying loads by having the average load slightly above rather than below the rated load of the motor, and this also requires a heat overload capacity. American industry cannot afford to be hampered by poorly designed motors. It demands motors that will stay on the job 24 hours a day and 365 days a year.

Cutting Proper Pipe Threads

Many threaded pipe joints are leaky simply because the threading has not been properly performed. A thread may be ruined in cutting by using a dull or improperly made die, or by forcing the cutting without supplying sufficient lubrication, the result being a torn thread that will probably leak when the piping is connected up.

A catalog entitled "National Modern Welded Pipe," published by the National Tube Co., contains some instructive information regarding correct threading principles, which is in part as follows:

It is highly important that threading dies be kept in good working condition, for even with uniform pipe it is difficult to secure good threads if the chasers of a die are

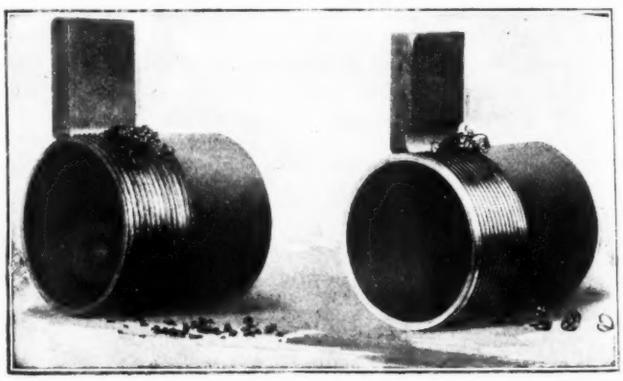


FIG. 1. ROUGH THREADS AND IMPROPERLY DESIGNED DIE

FIG. 2. CLEAN, SMOOTH THREADS AND PROPER DIE

lacking in proper lip angle, clearance in the lead or throat, have broken teeth, or if the die is lacking in chip space, sufficient number of chasers, etc.

For cutting threads on bessemer steel pipe, each chaser should have a lip angle of 15 to 20 deg., and for open-hearth steel pipe at least 25 deg. By grinding a slightly curved lip of this angle, an even cutting action is given to the chaser, similar to that of a properly ground lathe tool, and the effect of pushing the metal instead of cutting it is avoided, as shown by Figs. 1 and 2. Fig. 1 shows rough threads cut with an improperly designed die which causes the chips to crumble. Fig. 2 shows a properly ground

chaser which produces a clean, smooth thread and cutting spiral chips. If there is a square corner or shoulder at the top of the lip, this should be removed, as it forms a place where chips may lodge and pile up, resulting in torn threads and unnecessary friction.

Clearance is the space between the pipe threads and the teeth of the chaser. It is necessary to have proper clearance to prevent the teeth of the chaser from dragging on the threads of the pipe and spoiling them, and to give an easy cutting action to the die. The clearance should be a little greater in the lead of a die than at any other place, as this is the point of heaviest duty, but it should not be

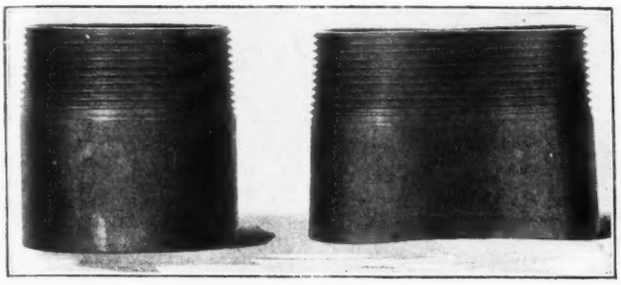


FIG. 3. THREADS OBTAINED BY USING PROPER DIE

excessive, as too great a clearance will produce results as bad as if no clearance were allowed.

Lead is the angle that is machined or ground on the first three threads (more or less) of each chaser to enable the die to start on the pipe, and also to distribute the work of making the first cut over a number of threads. The lead may be machined on or, as is more frequent, it may be ground on after the chasers are tempered. The proper amount of lead is about three threads. As the heaviest cutting is done by the lead, it should have slightly greater clearance angle than the rest of the thread on the chaser, but care must be used to see that this angle is not excessive. Too much clearance here will cause the die to lead too fast, and the half-threads cut by the lead are consequently damaged by the small teeth of the chaser. The illustration, Fig. 3, shows the result of a properly designed die which gives a strong thread that retains its shape when the pipe is flattened as in the right-hand view..

Improved Shape Suggested for Condenser Tubes

By L. Cammen

In view of the importance of surface condensers in power-plant engineering, it is rather surprising that inventors have given so little attention to this subject. There have been some improvements made in recent years, particularly in baffling in the steam space and better distribution of the tubes, but essentially the surface condenser of today differs little from its grandaddy of some sixty years ago.

This is largely due to the fact that the surface condenser apart from its auxiliaries (especially pumps, in which great advances have been made), seems to be an extremely simple piece of apparatus. It is essentially a casing containing a number of tubes, with steam flowing through the casing and water through the tubes.

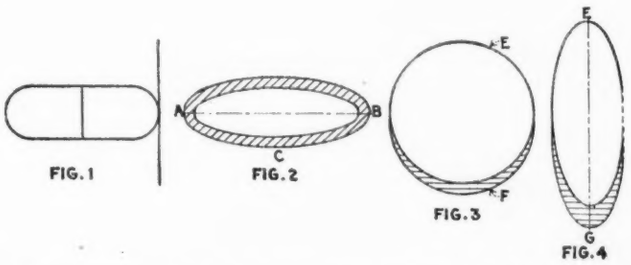
It is only natural to find tubes of circular section in universal use in surface condensers, for two reasons: First, round tubes are the cheapest and easiest to make, and are in fact the only kind of tubing made commercially; and second, round tubes are the only ones that can be made fast, without excessive tendency to leakage, in the condenser sheets. Therefore, round tubes have been used, without much regard as to whether they are the most efficient kind for this particular kind of service.

The efficiency of a condenser depends primarily on two factors: First, the degree to which the water flowing through the tubes comes in contact with the walls of the tubes, or useful contact; and second, the rate of heat transmission from steam to the walls of the tubes. In its turn useful contact depends on the amount of turbulence in the flow of water through the tubes and also on the size of the tubes. Water is a poor conductor of heat, and the water flowing through the center of the tube is of no practical use for condensation purposes unless it comes in contact with the walls of the tube or mixes with the water that does.

It is obvious that the greater the turbulence the more chance the center core has to be disturbed and brought into contact with the walls or outside layers. Also, the smaller the diameter of the tube the more chance there is for turbulence; but the tubes cannot be made smaller than a certain size, because they would have a tendency to clog up too soon with deposits from the water.

With regard to turbulence, there are only two ways which have been tried to produce it: First, by placing obstructions in the path of water flow in the tube (spirally wound ribbons, triangular metal blocks, etc.), and second, by increasing the velocity of flow of water through the tube. The first method is impractical, as on one hand it tends to increase the scale deposits in the tubes, and on the other hand it makes cleaning the tubes a very hard job. The second method is widely used now (the average velocities of flow through condenser tubes have increased in the last ten years not less than 100 per cent) and is giving good results, at a certain price in terms of additional pump equipment and power consumption.

The circular tube is the type in which there is the least tendency toward a turbulent flow, which is the



SHOWING SHAPE OF TUBES AND TENDENCY OF WATER TO CLING TO THEM

reason why circular tubes have proved so satisfactory for the transmission of liquids. This makes them, however, the least suitable for use where turbulence is desirable. It became obvious, therefore, to the writer that a section other than circular would have to be employed in condenser tubing in order to increase the useful contact between the water and the metal walls of the tube by increasing the turbulence of flow.

The section shown in Fig. 1 was tried first, as being the simplest to make (merely squeeze a round tube in a vise or between rollers), but its mechanical strength is low and it did not prove entirely satisfactory in other respects. The section shown in Fig. 2 was then developed. This is lenticular in cross-section and gives an extremely high coefficient of useful contact between the water and the tube for the following reasons: The

average distance of each particle of water from the metal walls of the tube is less than one-half that in a circular-section tube of the same area of cross-section, as can be easily shown either analytically or graphically; and second, the resistance to flow in the various parts of the cross-section is extremely uneven, being very large near the ends of the section *A B* and comparatively small along the minor diameter *C*, and it is difference in resistance in the various parts of the section that creates turbulent flow.

When steam condenses, it settles as water on the tubes. But water is a very poor conductor of heat, and while the film is actually never of any great thickness, it undoubtedly affects the efficiency of the condenser to a material degree. No attempt has ever been made to determine this factor experimentally, partly because of the difficulty of doing it, but mainly because the information, even if available, would not have been of practical use, so long as only circular tubes have been used, as all such tubes are equally affected by the water-film formation.

Simple tests have shown, however, that lenticular tubes behave in a manner materially different from circular tubes. In the case of circular tubes *E*, Fig. 3, the tendency for the water film is to adhere, as shown at *F*, covering nearly half of the surface of the tube *E*. Furthermore, the character of adhesion of the water film to the tube is such that a fairly large drop is maintained before it falls off or is blown away by the steam. With lenticular tubes, Fig. 4, the tendency is for the drop to form at one of the sharp ends *G* and the drop itself is smaller as it does not have as much surface to which to adhere. This has been found in very simple tests, where the drops in both cases were measured by weighing.

As a result of this there is more metal exposed to the action of the steam in the lenticular tube than in the circular.

A tube of lenticular section cannot be attached to the condenser sheet as snugly as one of round section. Because of this, a tube has been developed[1] which begins and ends, for about two inches at each end, with a round section, but is lenticular for the rest of its length. This is also of advantage, as tests have shown that there is less resistance at the condenser sheet with round tubes than with any other shape.

The temperature difference between the water at the inlet to and outlet from the tubes has to be the same no matter what the shape of the tubes may be if the same vacuum is to be maintained. Since, however, the lenticular tube is more efficient in the way of heat absorption from the steam than the circular tube, the length of the tube may be less, or, which comes to the same thing, in many cases a single-pass condenser might be used where a two-pass has to be employed now.

Cleaning of lenticular tubes is somewhat more difficult than that of round tubes. This, however, is partly compensated by the fact that the tubes themselves are shorter. Also, the greater velocity of flow of water through the tubes, together with the greater turbulance, reduces somewhat the amount of deposits in the tubes.

Heating of the commutator on a direct-current machine may be caused by overload, by sparking at the brushes, by too high brush pressure, by lack of lubrication on the commutator, or by circulating currents, in case of lap-wound armatures the bearings of which have become worn.

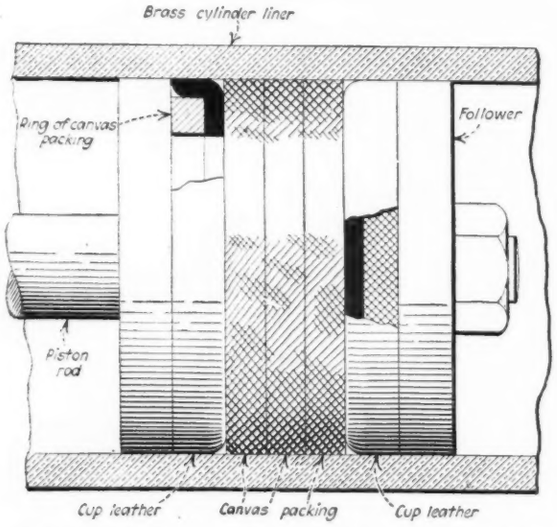
[1]Patent applied for.

Cup-Leather Packing for Pump Plungers

BY A. J. DIXON

A boiler is fed by a 5¼ and 3½ by 5-in. duplex steam pump. The pump draws water from the city main and delivers it, through a closed heater, to the boiler against a steam pressure of 130 lb. gage. The engineer in charge is a stickler for the small economies. He maintains that just as drops of water make oceans and grains of sand make mountains, so does the cumulative result of these seemingly insignificant economies amount in the course of time to a noticeable saving.

Originally, the water pistons of the feed pump were packed with square canvas packing. In order to run the pump at the proper speed for feeding the boiler—about 55 strokes a minute—it was necessary to open the globe valve in the steam-supply pipe one full turn. The engineer figured that if he could reduce the friction of



COMBINATION CUP LEATHER AND CANVAS PACKING

the packing in the water end, he could save something by further throttling the steam supply. Accordingly, he packed the pistons with a combination of canvas rings and leather cups, such as are used for packing the operating valves of hydraulic elevators. The job is shown in the sketch herewith. The canvas rings inside the leather cups are of ½-in. size; the three rings clamped between the cups are ⅝-in.

It is claimed by the engineer that the pump now supplies the boiler when the steam supply valve is open only one-half turn. He further claims that the pump does the same work as formerly with fewer strokes per minute. He attributes these advantages to diminished friction and to the elimination of water slippage between the packing and the cylinder lining.

Much significance is attached to the announcement by the Interstate Commerce Commission that it has cancelled service order No. 18, thereby taking away from the railroads the privilege of using assigned cars in the procurement of their own fuel coal, and that the whole matter of assigned cars is to be the subject of a formal hearing at a date that will be announced in the near future.

EDITORIALS

Engine Efficiency

EVERY pound of water in a mill pond one thousand feet above sea level has in it one thousand foot-pounds of potential energy relatively to that base. But if the lay of the land is such that a fall of only one hundred feet is available at one place, the rest being taken up by a gradual gradient to the sea, an ideally perfect turbine could convert into power only one hundred foot-pounds for each pound of water passed through it. If it did so convert eighty foot-pounds for every pound of the falling water, it would be said to have an efficiency of eighty per cent.

A similar condition exists with regard to the heat engine. It generates power by working heat from a higher to a lower temperature level—in the case of the steam engine from the boiler to the condenser or exhaust temperature. The "thermal efficiency" of the engine is the amount of heat converted by it into work, divided by the total amount of heat furnished to it above the absolute zero of temperature, analogous to the efficiency of the water turbine referred to sea level; but the measure of its real efficiency as a heat engine is the proportion which it converts into power of the energy convertible by working the heat furnished to it through the available temperature fall.

What engineers are interested in is not the efficiency referred to sea level or to absolute zero, but to the available range, the ratio of the work produced by the turbine or the engine to the energy derivable from the fall of the water through the *available head* or of the heat through the *available temperature range*. In the case of the water turbine this energy is derivable in only one way, but in the case of the heat engine either of several cycles may be used. If the heat were all put into the system at the higher temperature and all the heat that was rejected went out at the lower temperature, the cycle would be the Carnot, and the heat convertible would be proportional to the temperature drop, just as in a turbine where the water all goes in at the higher head and is rejected at the lower level, the energy derivable is directly proportional to the difference in levels. But in the steam engine the heat is not all put in at the higher level. It is received first by the feed water, usually at a comparatively low temperature and along a rising gradient until the temperature due to the pressure is reached, then at a constant temperature level during evaporation, then along another rising gradient during the period of superheating. And it is the energy derivable from the heat put into the system in these ways and reduced by adiabatic expansion to the lower temperature level with which engineers usually wish to compare the energy actually produced by an engine as a measure of its efficiency.

This cycle of varying temperature intake and adiabatic expansion to the lower limit came to be known as the Rankine cycle, and engineers came to call the ratio of the heat equivalent of the work produced by an engine per pound of steam to the heat convertible by working a pound of steam between the given limits of temperature and condition on the Rankine cycle as the "Rankine efficiency" of the engine.

And then came the purists, who told them that what they really meant by this was the Rankine cycle efficiency of the engine and that this was liable to confusion with the Rankine cycle efficiency or the efficiency of the Rankine cycle; that is, to the proportion of the total heat furnished above absolute zero which is convertible by an ideally perfect engine working on the Rankine cycle. And it was recommended that they call their quantity the "Rankine efficiency ratio" because it is the ratio of the thermal efficiency of the engine to the efficiency of the Rankine cycle.

And so they did—at least in print—and it is so defined in some of the textbooks and in the 1915 edition of the "Power Test Codes of the American Society of Mechanical Engineers."

And now in the process of revision of the Codes, Charles H. Smoot, a member of the committee of revision, raises an interesting discussion by pointing out that for a given case the denominators of the two efficiencies are the same; that when they are made into a "ratio," or one is divided by the other, the denominators cancel out, leaving not an "efficiency ratio" but a plain efficiency, a ratio of heat converted into power to convertible heat furnished, and urging that some simpler and more logical name such as "engine efficiency" be used for it.

Mr. Smoot appears to have a logical argument and a good case. The engineers who use the term are thinking of the relation of what they have actually produced to what they might have produced if their engine had been better designed or more effectively operated. They arrive at the values by measuring what the engine does and computing the energy derivable from the heat fall on the Rankine cycle or measuring it from a Mollier diagram—and do not think or figure around to it through thermal and cycle efficiencies. Most of its practical users would need to have the significance of "efficiency ratio" in this connection explained to them.

There ought to be a simpler and more logical name for "it" than "Rankine efficiency ratio." "Engine efficiency" is rather indefinite—the engine has many efficiencies. "Machine efficiency" is liable to confusion with "mechanical efficiency," commonly applied to the ratio of brake to indicated horsepower. Either would do as a general expression for the ratio of the heat converted into work to the heat so convertible, but for a specific case should have some qualifying term indicating the process by or cycle upon which the heat convertible is computed. In the meantime engineers will go on calling it the Rankine efficiency, each understanding what the others mean by it, and never tangling the Rankine efficiency of their engines and turbines up with the thermal efficiency of the Rankine cycle, the quantitative value of which is so different as to make any confusion in either practical use or abstract thinking impossible.

Rational Design of Power Plants

IN THE DESIGN of power plants the problem may be summarized as producing and delivering power to the feeder system or transmission lines outside the plant at a minimum cost per kilowatt-hour, including fixed charges, consistent with reliability, safety and continuity of service. The cost of energy at the busbars has frequently been referred to as the criterion. However, when power is delivered to the transmission system through step-up transformers at the station, the question of the most economical location for the transformers—whether to place them in the power station or outdoors—becomes part of the problem if the most economical results are to be obtained. Low flue-gas temperatures, high CO_2, high boiler ratings, high steam temperatures and pressures, and low water rates in the turbines are all regarded as desirable features; but a plant might be designed to obtain all of these and yet not deliver a kilowatt-hour to the transmission system at minimum cost, because of the increased overhead and maintenance.

The leading article in this issue describes the Colfax station of the Duquesne Light Company, of Pittsburgh. In the design of this plant there are many features that might be interpreted as being in contradiction to past practices. Steam temperature of less than six hundred degrees Fahrenheit is used, economizers have not been installed, and the boilers are to be operated around two hundred per cent rating, etc. However, experience with high-temperature steam has led many to feel that six hundred degrees is about the maximum desirable, with the present state of the art. If a boiler plant is designed to obtain the highest thermal efficiency, then economizers are justified; on the other hand, if the end sought is to deliver a pound of steam at the turbine throttle at minimum cost, the cost of the coal saved by their installation must be weighed against the additional overhead and maintenance charges entailed by this equipment. The Colfax station is situated about a mile from the mine's mouth, consequently the cost of coal delivered at the plant is low and probably will be for many years to come, and this fact influenced the decision against the use of economizers.

The foregoing is true of many other features of this station; they are not radical departures in power plant design, but have been worked out on the basis of cheap fuel and other existing conditions, so as to deliver a kilowatt-hour to the transmission system at the lowest cost, including fixed charges, consistent with reliability. Of particular interest are the boilers because of their large capacity and high settings and the fact that they are fired under the low end. The results obtained with the exceptionally large furnace volume per square foot of heating surface will be watched not only from the combustion viewpoint but also as to maintenance of the furnace. Although the location of the plant is such that cinders will not be a nuisance, nevertheless the large furnace volume will afford an opportunity for studying this feature as a means of eliminating cinders, which have caused so much trouble for plants situated in cities. This plant is the first one containing such large turbine units, in operation, that has been designed from the ground up. Therefore the results obtained will be of particular interest since the design is not only conservative but simple throughout.

Superheat with Variable Load

EMPLOYMENT of superheat has closely followed the growth and application of the steam turbine, because of its influence upon the efficiency of the latter. However, when laying out a turbine plant it is not unusual for the designing engineer to concern himself with the steam capacity and maximum temperature without paying attention to the load conditions in so far as they are likely to affect the degree of superheat; that is, the turbine is expected to show a predetermined performance at a given load, pressure, vacuum and superheat, and it is taken for granted that these conditions will be attained. To control the first three is not difficult, but to maintain a constant superheat at the turbine throttle under varying operating conditions is a problem that has taxed the skill of the superheater manufacturer and the patience of the operators.

The degree of superheat is dependent principally upon the ratio of the amount, velocity and temperature of the products of combustion sweeping past the superheater, to the weight of steam passing through it. This ratio, as pointed out by Mr. Pratt elsewhere in this issue, is dependent, first, upon the fuel burned; second, upon the combustion conditions; and third, upon the variation in load. Of course, in the final analysis it all reverts to the furnace conditions. If it were possible under varying load and with different fuels to maintain a constant percentage of excess air, CO_2 and gas temperature—in other words, a flat boiler-efficiency curve—one would then expect a constant superheat; but this is not possible. The best that can be done is to acquaint the superheater manufacturer with the properties of the fuel to be burned, the furnace conditions aimed at and the probable load variations. He will do his best, but too much should not be expected of him.

This variation of superheat with load sometimes means wet steam or slugs of water at the turbine, resulting in temperature shocks that are injurious to the metal and set up destructive stresses in the blade fits, rings and wheels. That it is possible for water and superheated steam to exist together without being apparent from the thermometer reading further complicates the matter, but it is generally conceded that careful attention to the water level under varying load will at least help.

In a recent Foreword on "Human Engineering in the Plant," W. A. Darter hit the nail on the head when he said that a man that fails to train another to fill his place stands a poor chance of promotion because of that very fact. Some men strive to become efficient at their work and then make believe that no one else could fill the job one-half as well. Canny executives do not encourage such an attitude, but neither do they as a rule actively oppose it, for it is not especially bad business from their viewpoint. For the man, however, it is likely to mean stagnation.

A fortunate coincidence is the selection of the same city and practically the same date for two big conventions this year. Jot down in your memorandum book that the American Society of Mechanical Engineers and the National Electric Light Association will hold big conventions at Chicago from May 23 to 26 and May 31 to June 3 respectively. The four-day interval will provide an opportunity for some sightseeing and relaxation.

CORRESPONDENCE

Repairing a Fractured Pump Cylinder

The water cylinder of a large compound direct-acting pumping engine, after giving many years of uninterrupted service, began to crack here and there. The fractures were probably due to long-continued straining in the grain of the iron, caused by a certain undulatory motion or breathing induced by the reciprocating motion of the plunger. Perhaps foundation instability might have had something to do with it. At any rate the appearance of the contact between the base plate of the cylinder and the grouting underneath indicated that a slight teetering action might have occurred.

At about the point *A* in Fig. 1, the first crack appeared in the cylinder. This crack lengthened in both directions until it had extended to a distance of about 22 in. There was considerable leakage from it when the pump was running. It was repaired by first chipping a tapering groove, wider at the bottom than at the top, throughout the length of the crack and driving in a strip of lead. A $\frac{5}{8}$-in. steel patch-plate was then trimmed to the proper size, and a row of $\frac{7}{16}$-in. holes, 1 in. apart, were drilled entirely around next to its edge. This plate was bent to the curvature of the cylinder. It was next placed centrally over the crack, and the holes were laid off on the cylinder casting. The holes were then drilled and tapped in the cylinder to receive $\frac{3}{8}$-in. capscrews and the patch, with a gasket of $\frac{1}{16}$-in. sheet lead underneath, was fastened on, as indicated in Fig. 2.

This job was no sooner completed than a second crack, of about the same length and at about the same height as the first one, but somewhat nearer the middle of the cylinder, appeared in the opposite side of the cylinder wall. This crack was repaired as the first one had been. But as a means of reinforcement, a massive U-bolt, *C* Fig. 2, was forged and yoked around the cylinder fore and aft, just beneath the flange to which the discharge-valve deck is bolted.

With these trimmings annexed, the pump ran awhile without showing any further signs of weakness. But eventually, a third crack was noticed, starting at about the point *B* below the suction-valve deck, as indicated in Fig. 1. In a little while this crack was about 24 in. long. Before it could be repaired, however, a companion crack, of about the same length and almost exactly opposite on the other side of the cylinder, was discovered. Both cracks were grooved, calked and patched, as shown in Fig. 2, just as the others had been.

As is also indicated in Fig. 2, this repair was supplemented with a reinforcement consisting of heavy wrought-iron straps laid across each end of the cylinder and joined together with tierods along the sides as at *D*. A series of holes were drilled in the cast ribs or buttresses between the cylinder wall and the baseplate, and the tierods were slipped through.

The pump now continued to run, without showing further signs of rupture, for about five weeks. It was then noticed that considerable water was running from beneath the wrought-iron clamping strap that traversed the outer end of the cylinder beneath the head. When the clamp was taken off, the cylinder was found to be cracked straight across, as shown at *E*, Fig. 3. This crack was grooved and calked with lead as the others had been, but no patch was bolted on. Instead, the face of the heavy wrought-iron strap was fitted nicely to the

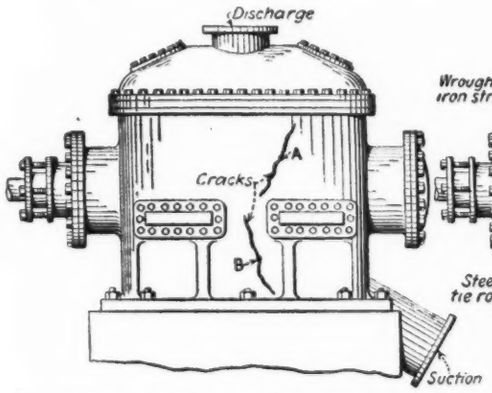


FIG. 1. CRACKS IN CYLINDER WALL OF PUMPING ENGINE

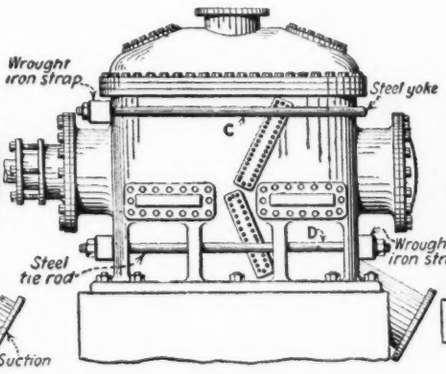


FIG. 2. SHOWING CYLINDER WHEN PATCHED AND BRACED

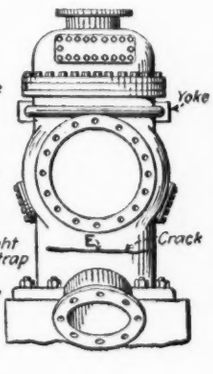


FIG. 3. CRACK AT END OF CYLINDER

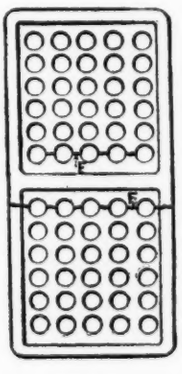


FIG. 4. CRACKS IN SUCTION VALVE DECK

surface of the cylinder casting, a gasket of $\frac{1}{16}$-in. sheet lead was inserted, and the clamp was drawn up.

But this did not end the trouble. For some time it had been noticed that the loss of pumping effect, generally attributed to slippage, was excessive. One day, while exploring the suction-valve deck in a search for leaky valves, the engineer discovered a fissure extending from hole to hole across the deck on each side of the middle partition, as indicated at *F*, Fig. 4. In the

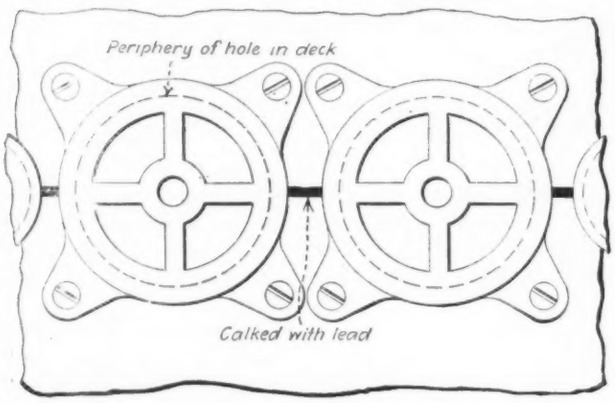


FIG. 5. SPECIAL SEAT BUSHINGS FOR VALVE DECK

chamber at the outer end the fissure appeared to join with the cracks that had previously shown in the external wall. But in the chamber at the inner end the fissure did not extend beyond the valve deck. The slippage through the crack in the inner-end chamber was comparatively slight, since the edges of the fracture at this end abutted closely. But in the outer-end chamber the fracture gaped open to an extent that permitted the seat bushings to flop around in the holes.

It was difficult work to groove and calk these cracks. But with about two days steady effort the job was finally accomplished. It was necessary, however, to design special seat bushings for the distorted and enlarged holes in the valve deck of the outer-end chamber. These bushings were cast with four projecting ears, by which they were fastened to the valve deck with countersunk screws, as indicated in Fig. 5.

The pump has now been running for some time with the patched and trussed-up water cylinder in as serviceable condition as it ever was. The original cause of the trouble, the undulatory motion of the cylinder, has apparently disappeared. As a precaution against its recurrence, however, the base of the cylinder has been regrouted.

A. J. DIXON.

St. Louis, Mo.

Pounds of Steam Produced

I wish to call attention to a statement in the issue of March 15, page 418, under the sub-heading "Total Pounds of Steam Produced," wherein it is stated that this is the "total amount of feed water for a given time as shown by the meter."

This can be true only when the boilers are not blown down and when there are no leaks in the feed-water line. During a trial test extreme care is taken to see that the blowoff valve is not opened, but for continuous operation the boilers must be blown down. I have found that a favorite although unscrupulous way of obtaining fictitious values of evaporation per pound of coal is to allow the boiler blowoff valves to remain partly open and cause the feed-water meter to register higher than it should. Obviously, if one takes the feed-water meter readings at their face value, it is easy to obtain high evaporation figures, and one does not need to worry the firemen or anyone else about the fires or CO_2 tests. As a matter of fact, I have seen this thing carried so far that records were turned in showing evaporations of 14 and 16 lb. of water per pound of coal.

I would, therefore, like to suggest that it will be more accurate to state that the pounds of steam produced are equal to the water fed into the boilers, as shown by the feed-water meter, less the blowoff and leakage losses.

C. G. HADLEY.

Southbridge, Mass.

Casting of Flywheels

Many lives have been lost and much property has been destroyed by the bursting of flywheels, from one cause or another. Contraction strains are bad. I believe that one way to avoid them would be to use steel spokes, with both ends upset, or a hole made in the ends with a drift pin when the arms are hot, so that the metal would run around the upset or into the holes while being cast. The spokes are to be placed in the mold and the wheel rim cast. After this casting has cooled, the hub of the wheel should be cast around the inner ends of the arms. When this casting cools, there will be a tensile strain on the arms in the right direction.

I do not know whether there is steel obtainable of the proper shape or not, but the steel mills would make it if there was a demand.

J. S. REDMOND.

New York City.

Redesigned Boiler Furnace Reduced Coal Consumption

Redesigning the furnace and combustion chambers of a 48-in. by 14-ft. horizontal return-tubular boiler produced an increase in efficiency from 42 to 61 per

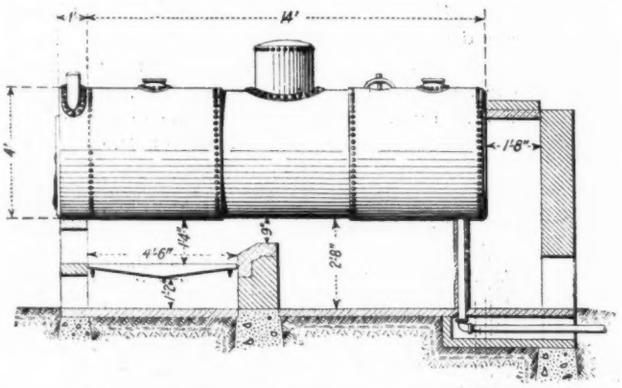


FIG. 1. ORIGINAL FURNACE AND SETTING

cent, or 19 per cent, and relieved the manufacturer from installing a new unit.

Before making any changes in the boiler setting, Fig. 1, a test was made of this furnace and boiler, which showed a high percentage of fuel waste. It was not generating enough steam to keep up the production of the plant.

After making the changes, as shown in Fig. 2, the fuel consumption was reduced between 1 to 1½ tons a day. This furnace and boiler, from Nov. 22, 1920,

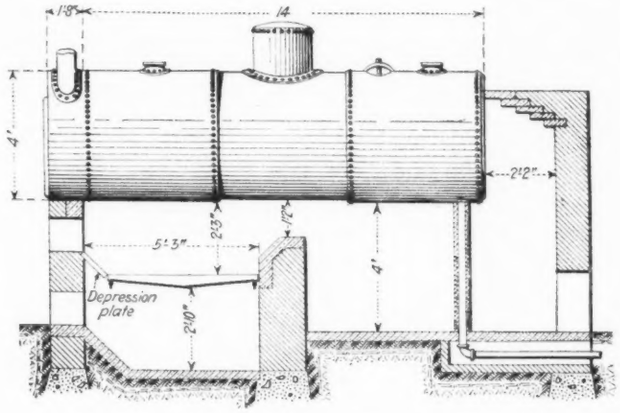


FIG. 2. CHANGED FURNACE AND SETTING

to Feb. 22, 1921, showed a saving of 135 tons of coal, and the steam generated was in excess of the load requirements. C. J. MILLER.

North Tonawanda, N. Y.

[It is evident from the foregoing that the increase in furnace volume had considerable to do with the saving made in coal consumption.—Editor.]

Are Small Boiler Tubes Desirable?

The discussion regarding small boiler tubes that has been going on for several issues is quite interesting, but it does not seem to get anywhere, and a few remarks from an old timer, whose "hat is still in the ring," though possibly not among the heavy weights, may not be out of place.

Way back in 1874, in company with the late Prof. R. H. Thurston and John B. Root, I had much to do with formulating a code for testing the efficiency of steam boilers, which later became the foundation for the code at present in use as recommended by the A. S. M. E.

During all these years I have made many hundreds of evaporative tests under greatly varying conditions, and it is my belief that the size of tubes within the limits used in the construction of the common horizontal return-tubular boiler has little to do with its efficiency. Is it not more a question of getting the proper size of tube for the kind of fuel to be used? It is a well-established fact that, other factors being the same, 4-in. tubes in a shell 18 ft. long are best for wood or bituminous coal, whereas 3-in. tubes under like conditions are equally as good for any short-flame fuel.

And yet the fact remains that the highest efficiency I ever obtained, and probably the highest ever obtained from any boiler in actual every-day service over a period of several years, was secured from a vertical tubular boiler of about 3,000 sq.ft. of heating surface, and fitted with 2½-in. tubes, 18 ft. long and fired with bituminous coal.

When it is considered by many that the vertical tubular type of boiler is not remarkable for high efficiency, these facts seem to refute a whole lot of theories, and possibly it is a question of proper spacing of tubes and provision for circulation that affects the capacity and economy of a boiler of the horizontal tubular type, more than the size of the tubes, for, as we all know, many boilers of this type are bought on the basis of heating surface for a given price rather than on a guaranteed efficiency of the surface, and this leads some boilermakers to crowd an undue number of tubes in a given shell.

While on this question of steam boilers, is it not a fact that high duty depends more on the human element in charge of the fireroom than it does on the design of the boiler itself? Then again, is it not a fact that many owners of small jobbing shops who can turn out a good workman-like job know very little about what might be called the science of steam generation?

What leads to this latter question is that only recently a shop owner asserted in my presence it was only the lower rows of tubes that did efficient work, whereas in the late sixties when I had to dig my facts out for myself because there were no such textbooks as we have today, and demonstrated to my own satisfaction that the reverse is the case. This demonstration was made by placing strips of pine wood in the back end of the tubes about three feet from the tube sheet. The upper strips were burned out the first day, but it was several days before those in the lower tubes were even badly charred.

The moral of all this is, When your watch needs repairs take it to a watchmaker; when your shoes need repairs take them to a shoemaker, but do not reverse this process. When it comes to a mechanical question, take it to a reputable mechanical engineer, and be sure you are right before you go ahead. W. H. ODELL.

Brooklyn, N. Y.

Fitting Diesel Piston Rings

Referring to the article in *Power*, March 8, 1921, by P. P. Talatz, on Diesel piston rings, I think he is in serious error in assuming that an elliptical ring will fit the cylinder walls any more than would a circular ring unless the former be fitted by scraping.

Piston rings are turned with the inside and outside eccentric to give them more flexibility and more spring. However, even this type needs to be fitted by scraping, etc. The editor's suggestion of scraping was one to be followed. J. R. PETRIE.

San Francisco, Cal.

Starting Up a Steam-Turbine Plant

In reading W. Aspril's article in *Power*, page 144, Jan. 25, I was forced to smile. Mr. Aspril's "dope" is good, and reading between the lines I assume that he has a job where he is boss, and a job that he is absolutely sure of. Unfortunately, it is not that way with all of us. I have worked in plants where his system would have meant a new chief in a hurry. I take it that he has never had charge of a steam stand-by station for a big hydro system, or he would find that getting on the line in this sort of plant is a business in itself, calling for speed and nerve, and an abundance of both.

Some time ago I was in charge of a steam stand-by station in which there were two 5,000-kw. turbines and ten boilers. Our incoming current was about 1,500,000 kw.-hr. to be distributed per twenty-four hours. Under normal conditions we ran with our own plant cold,

as the periods of service with us were often four to five weeks apart.

In the event of loss of load in one of our hydro plants, due to breakdowns, etc., we were supposed to get on the line and make up whatever reduction had been made in our incoming power. The following incident will illustrate the method we follow in answering a quick call.

A call comes in at 6 p.m. that a heavy load is coming on and that the flashboards on No. 4 hydro plant are gone. Pick up the whole load of this station, about 8,000 kw. The horn is sounded by the operator calling all boiler-room help to the boiler room. No man was supposed to stop for any cause, and I have seen men off watch who answered the call, half shaved or only partly dressed. Fires on the grate are hand-laid, shavings oil soaked, and the wood and hand-picked coal are dipped in oil. Forced-draft and induced-draft fans are motor-driven, as are the stokers. Current is taken from the main distribution bus, thus providing against any station loss in the system. Unless every plant in the system goes flat, we can get "juice" to our motors. The fireman starts the fire, the helper starts the stokers and fans, and the engineer looks over the valves. I have personally brought the steam from zero up to 200 lb. gage in forty minutes, and at the end of the forty minutes we had a 32-in. fire on the grates, the side doors were white hot, and the dump plates were melting.

No attempt was made to get the turbines ready until the steam reached 200 lb. gage, as getting a machine on the line without steam enough to carry an overload, might result in another shutdown. Once we get on the line, we have got to stay there; as soon as the damaged machines are off, we are supposed to carry the load; our failure will mean a slowing down of the system, reduction in frequency, and the added difficulties in getting back on the system.

With 200 lb. pressure on the steam gages, we start the pumps on the circulators, air lines and oil system, this work being divided between a turbine operator and two oilers. When the vacuum reaches 26 in., the switchboard operator closes the throttle switch, and the turbine begins to come up to speed. Fields are closed, synchronoscopes are plugged in, and on the first turn of the pointer the line is closed. From a standing still, we have put a turbine on the line with a full load in two minutes and eight seconds.

Bear in mind that where we are losing a big mill load our instructions are to give no consideration to our equipment. I don't pretend to say that this is good practice, but it is the kind of job that I like; it makes me feel a man to know that I can swing these giants to work under bad condition without a smashup. In other words, I like to take a chance.

We furnish "juice" to big mills having contracts which make us a heavy loser when we fail to come through with the power. Hence, it is our business not to curtail our customer's production; this is what I preached to my crew and I had their co-operation to a man. My motto is "To hell with my machines"; if my people want power it's my job to give it to them, and when my machines are gone they can supply me with new ones.

In winding up, I should like to tell of the attitude one of my bosses took when I ran my plant on the policy stated. We got a hurry-up call to pick up a 10,000-kw. load, and in 44 minutes we were on the line, but at the end of six hours' run I had two ruined boilers on my hands without stokers or settings. I figured, being young at the game then, that I was finished as an engineer, but the next morning my old superintendent came into the plant with a smile and a cigar, and giving me a slap on the back remarked: "You done well, my boy; we can put two new boilers in for $30,000, and if you had failed me last night we would have lost twice that in damages, and you would have lost a good job."

A. D. PALMER.

South Quincy, Mass.

Allowable Temperature Rise for Motors

In the Jan. 11 issue of *Power* Mr. Phillips has brought up a subject of vital interest to all power users and especially so to those of the Southwest. Several of the larger copper companies expect to maintain their present standard specifications and will continue to call for 35-deg. rise motors for their use. Some are expecting to specify altitude, temperature and humidity and allow a rise of 50 deg. only at their actual running conditions.

A great deal has been said regarding the increased efficiency and power factor of the 50-deg. motor, but little has been heard from the actual user. It would be folly to belittle the efforts put forth in perfecting this motor. This has really been a notable achievement, but to try to force it as an absolute standard in all cases would not appear advisable. Many adverse criticisms have been made on the quality of the motors turned out by some of the manufacturers because the 50-deg. motor was sold and placed in service for which it was unsuited.

As applied to a mining or industrial process, reliability in operation will far outweigh a small saving in motor efficiency or power factor. The real efficiency to be considered is the efficiency of the process and not the individual efficiency of the component machines. If a motor saves 2 per cent in the use of power but increases delays due to motor failures by 1 per cent, the net loss in the first shutdown may be more than the power consumed in a year and the cost of the motor combined. Thus the most efficient motor electrically may be the least efficient from the viewpoint of the process. Another thing that should be considered is the cost of maintaining the motor in service. As a rule the increased supervision required will outweigh the saving in power.

In many cases motors are bought under a misconception by the superintendent or purchasing agent, to whom this difference is represented as a highly technical matter of little importance from an operating point of view. After the motor is installed, the plant electrician is expected to get the same service from it as from the older 40-deg. motors. If he cannot, he is incompetent, as the motors were made and sold by a large manufacturer. Both are dissatisfied, and in many cases do not know the real reason.

In the Southwest we frequently have room temperatures of 125 to 130 deg. F. in the summer. While actual charring of the insulation may not occur below 117 deg. C., a motor operated at an average temperature between 90 and 100 deg. C. will show some damage to insulation after a run of two or three months, probably due to hot spots. With a room temperature of 50 deg. C. and a normal rise of 50 deg. C. at full load, the temperature is within striking distance of the danger point.

Another factor to consider is, the motor cannot dissipate its heat at high altitudes as well as at sea level. Owing to the lower humidity and higher altitude the rise at full load will be increased from 3 to 6 deg. Under these conditions the 40-deg. motor cannot carry much overload without undue deterioration and the 50-deg. motor cannot be run at full load.

From the viewpoint of the average mine chief electrician there is considerable humor expressed in the statement, "The motor should be installed in a clean, dry place." Respirators or hip boots are frequently worn by the gangs in the actual installation work. Under conditions such as these the efficiency can well be disregarded for reliability. The newer motors depend considerably upon ventilation through small openings in the rotor and stator to dissipate the heat. This may work well in some industries, but in mining and some of the heavy industries they are usually closed with dust or other material in the first month or two of operation. To clean them thoroughly necessitates a complete dismantling of the apparatus, requiring a considerable increase in operating labor, which will more than wipe out the power saving. Around a crushing plant or any industry where there is much dust, the motors cannot readily dissipate their heat, and we frequently cannot expect more than normal full load from the 40-deg. motor without reaching a dangerous temperature.

There are times in these industries when the class of labor handling the apparatus is such that the machines must pull overloads or seriously interfere with the process on account of carelessness or negligence. This will frequently justify overfusing and dependence on the motors to carry the load. It is often the case that the small elevator or conveyor motors are really of more importance than the large motors on the individual mills. A shutdown in one of these is a serious matter, and they are usually fused in such a manner that only serious damage to the motor or wiring will blow the fuse. Since the advent of the 50-deg. motor the practice of installing duplicate motors is growing. In case a motor goes out, the changing of a belt will enable resumption of operation. This insures practically continuous operation, but at a considerably increased installation cost.

The overload requirements of this motor are too rigid for an isolated section where motors are frequently ordered before the definite selection of the machinery. Load conditions are changed with no motors of exact size on the job and none available for months. For these conditions considerable leeway in overload is desirable. The first cost of a satisfactory machine is soon forgotten, but the unsatisfactory machine is shifted to an unimportant duty and kept there.

It has been argued that transformer ratings have been raised to 55 deg. In this we have an entirely different set of conditions. The windings are submerged in oil and are kept clean; the oil prevents the insulation from drying out, and the circulation readily removes the heat from the windings and cores.

If conditions are such that the power factor must be brought up to a fixed standard, it is far better to install a static or synchronous condenser for that purpose than to endanger reliability or continuity of operations.

No doubt the 50-deg. motor is here to stay. It must have its place or it would not have come. It is very difficult to convince a superintendent or general manager that in order to carry a load of 30-hp. in a satisfactory manner it is necessary to purchase a 50-hp. motor. That is what it virtually means for many conditions, and the 50-hp. will cost more than the old 30-hp. If we must have the 50-deg. motor why not use an asbestos or mica insulation and thus make the winding of a class which will withstand the heat?

T. H. ARNOLD.

El Paso, Tex.

Standard Unit of Refrigeration

In an exhaustive article giving the history of the recently adopted Standard for Refrigerating Capacity, appearing in *Power*, Feb. 1, 1921, the conclusion arrived at, in the last paragraph of the article, is that "under the adopted standard the rating of refrigerating machines will be about 1.3 per cent lighter than under the old suggested temperature levels." This reasoning is based on the fact that with 0 and 90 deg. F. ammonia temperatures (A. S. M. E. basis of 1904) the quantity of liquid ammonia to be evaporated per minute per ton refrigerating effect equals $\frac{200}{468.1}$ or 0.42726 lb., while under the temperatures 5 and 86 deg. F. adopted by A. S. R. E. in December, 1920, $\frac{200}{474.3}$, or 0.42167 lb. is required, which is 1.3 per cent less than before.

Applying this information to commercial work, one would be very likely to assume that a machine rated 100 tons refrigeration at 0 deg. evaporator temperature and 90 deg. condenser temperature will, with 5 and 86 deg. temperatures, develop $100 \times \frac{0.42726}{0.42167} = 101.326$ tons refrigeration.

One would naturally conclude that the new standard hardly warrants any change in the rating of existing machines.

This, however, is not the case, because the output of refrigerating machines, at least those of the compression system, depends on their ability to circulate given volumes, not weights, of refrigerating fluid.

At 0 deg. F. and 15.72 lb. gage pressure, using the table of properties of anhydrous ammonia issued by Bureau of Standards in December, 1920, and published in *Power*, Jan. 18, 1921, the volume per pound of vapor is 9.114 cu.ft. Hence a 100-ton machine would have to circulate,

$$100 \times 0.42726 \times 9.114 = 389.405 \text{ } \textit{cu.ft. of vapor per minute.}$$

At 5 deg. F. and 19.6 lb. gage pressure, the volume per pound is 8.149 cu.ft. In this case a 100-ton machine would have to circulate,

$$100 \times 0.42167 \times 8.149 = 343.619 \text{ cu.ft.,}$$

or the original 100-ton machine would develop

$$100 \times \frac{389.405}{343.619} = 113.4 \textit{ tons.}$$

One standard in almost universal use is that adopted by the York Manufacturing Co. and applied to its machines for probably twenty years now. It contemplates 0 deg. F. evaporator temperature (now equivalent to 15.72 lb. gage pressure) and 96 deg. F. liquefaction temperature (now equivalent to 184.2 lb. gage pressure). Under this York standard the weight of liquid to be circulated per minute per ton (assuming

as usual, no subcooling of liquid before reaching the expansion or feed valve) equals

$$\frac{200}{568.7 - (150.5 - 42.9)} = \frac{200}{461.1} = 0.43375 \; lb.$$

And since the volume per lb. of vapor is 9.114 cu.ft., a 100-ton York machine would have to circulate

$$100 \times 0.43375 \times 9.114 = 395.32 \; cu.ft. \; vapor.$$

Under the standard temperature, 5 and 86 deg., adopted by the American Society of Refrigerating Engineers, December, 1920, the 100-ton York machine would be doing

$$100 \times \frac{395.32}{343.619} = 115.046 \; tons.$$

So instead of about one per cent difference, the rating of most of the machines in operation today is increased by the new standard by 1.15, theoretically.

In practice the ratio is greater still, because of the easier working conditions allowed by the A. S. R. E. standard. Under the York standard (0 and 96 deg.) the ratio of compression within the cylinder is at least

$$\frac{184.2 + 14.7}{15.72 + 14.7} = \frac{198.9}{30.42} = 6.54 \; to \; 1$$

while under the A. S. R. E. standard the ratio of compression is

$$\frac{154.5 + 14.7}{19.58 + 14.7} = \frac{169.2}{34.28} = 4.94 \; to \; 1$$

so that with equal clearance volume the loss in pumping capacity due to re-expansion is correspondingly less. The lower ratio of compression results also in a lower discharge temperature, and thus the loss due to cylinder heating is reduced, the vapor admitted striking walls of lower temperature and therefore not expanding as much.

The York Manufacturing Co. prescribes for 0 and 96 deg. an effective piston displacement of 8,900 cu.in. or 5.153 cu.ft. per minute per ton (mean value for their single and double acting compressors). This corresponds to a true volumetric efficiency of $\frac{3.9532}{5.153} = 0.767$

With pressures corresponding to 5 and 86 deg. York Co.'s catalog gives a mean value of 7,300 cu.in. or 4.226 cu.ft. per ton, corresponding to a true volumetric efficiency of

$$\frac{3.4362}{4.226} = 0.8125.$$

These efficiency figures bear out the statements just made, that under the adopted A. S. R. E. temperatures a machine will have a higher real volumetric efficiency.

On the basis of the York displacements just given, the actual output of a machine working at 5 and 86 deg. compared to one working at 0 and 96 deg. is

$$\frac{8900}{7300} = 1.22 \text{ to } 1.$$

In other words, the adopted A. S. R. E. standard (O and 86 deg.) increases the capacity of ammonia compression machines heretofore rated at 0 and 96 deg. *theoretically* by 1.15, and *practically* by 1.22.

If the case of the so-called high-speed ammonia compressors, introduced since a few years, the increase in rated capacity may reach 30 to 60 per cent, because of the lesser re-expansion obtained with a low ratio of compression in cylinders having 3 to 6 per cent clearance volume. The figures given in the above examples refer to compressor cylinders having clearance volumes of less than 2 per cent.

The mere adoption of a rating based on standard conditions has no influence upon the actual output of machines forced to continue to operate under other than standard conditions. Where a greater output is desired a substantial improvement may be made by adopting such expedients as: larger condensing surface, thus lowering the condenser pressure; using larger evaporator surface to raise the suction pressure; supply more or colder condenser water, employ a water-using liquid forecooler; insert a liquid separator, so as to run dry compression; adopt low gas velocities in compressor valves and mains, and use ample and effective insulation.

It was, perhaps, an oversight that in the adoption of a standard the temperature of the liquid as it reaches the expansion valve was not specified. In European standards this is always taken into account. In a refrigerating system, especially where counter-current condensers or a separate liquid cooler is used, the condensed liquid can be cooled with water to a temperature lower than the liquefication temperature, and thus the yield of the machine increased, all other conditions remaining the same.

If under standard vapor temperatures, the liquid reaches the expansion valve at 66 deg. instead of 86 deg. the useful refrigeration per pound of vapor circulated is increased, because less cooling is wasted in lowering the temperature of the refrigerating agent to that of the evaporator. With 86 and 5 deg. the difference in heat content of liquid is 138.9 — 48.3, or 90.6 B.t.u.; with 66 and 5 deg. it is only 116 — 48.3, or 67.7 B.t.u. This leaves available for useful refrigeration, 564.9 — 67.7, or 497.2 B.t.u. per lb.; so that only $\frac{200}{497.2} =$ 0.4023 lb. liquid must be circulated by the machine per minute per ton refrigeration.

At 8.149 cu.ft. per lb., 3.278 cu.ft. of vapor must be circulated, against 3.4362 cu.ft. when liquid is not subcooled. A 100-ton machine of identical size and speed, and standard discharge and suction pressure and temperatures, would then perform $100 \times \frac{3.4362}{3.278} = 104.8$ tons.

Evidently, for every 4 degrees subcooling of condensed liquid, the output of the machine is increased by about 1 per cent, and since the horsepower remains as it was, the horsepower expended per ton of work done is diminished.

Hence, to obtain a true measure of the performance of a refrigerating machine, it is necessary to take into account the temperature of the refrigerating liquid at the entrance to the cooling coils or evaporator and not to stop at the high and low saturation temperatures of the vapor.

Among the minor corrections that may be made in the article cited for the benefit of those who desire exact data may be mentioned the following:

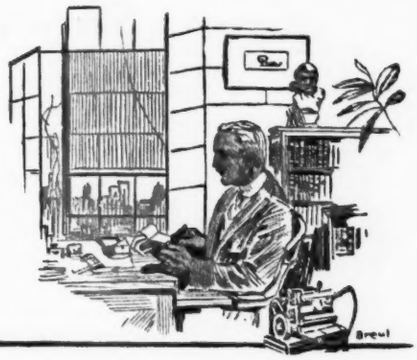
On page 185 of the Feb. 1 number the heat transmission given for the insulated wall as 0.00333 B.t.u. sq.ft., deg. F. difference, should read "per minute" and not "per hour." The equivalent of about 1½-in. thickness of pure corkboard will give this insulating effect.

On page 186 the 86-deg. liquefaction temperature in the case of Pictet fluid corresponds to 49.85 lb. gage pressure, not 6 in. of vacuum, this fluid consisting chiefly of SO_2. For methyl chloride it is 80.83 lb., not 8.083 lb.

CHARLES H. HERTER.

New York City.

INQUIRIES OF GENERAL INTEREST



Metallic Rod Packing for Old Compressor—Can metallic packing be used on an old ammonia compressor piston rod? R. A.

If the rod is not scored or shouldered, there is no reason why metallic packing suitable for ammonia should not be used.

Frosting of Suction Line—How far should the suction side of a refrigerating system be frosted back? H. S. C.

Most engineers find it best to carry the frost back to the suction-valve connection, if this does not make the discharge line too low in temperature.

Weight of Compressed Air Contained by Tank—What weight of air at an absolute pressure of 60 lb. per sq.in. and temperature of 90 deg. F. would be contained in a cylindrical tank 8 ft. in diameter and 20 ft. long, neglecting the volume of bumped heads? G. L.

The weight of air is given by the formula

$$W = P\,V \div 53.2\,T$$

in which

W = Weight in pounds;
P = Absolute pressure in pounds per square foot;
V = Volume in cubic feet;
T = Absolute temperature of the air.

The absolute pressure would be $60 \times 144 = 8{,}640$ lb. sq.ft., the volume of the air would be $8^2 \times 0.7854 \times 20 = 1005.3$ cu.ft., and the absolute temperature would be $90 + 460 = 550$ deg.; hence the weight $W = (8{,}640 \times 1{,}005.3) \div (53.2 \times 550) = 296.8$ lb.

Use of Too High or Too Low Indicator Spring—In using a steam-engine indicator, what happens when the scale of spring is too high; and when it is too low? H. P. M.

The higher the scale of spring the shorter the pressure ordinates of the diagram and less conspicuous the pressure changes and back pressure, and greater the percentage of error probable in making measurements of pressure and determination of the m.e.p. shown by the diagram. When the scale of the spring is too low, there may be waviness of the lines of the diagram from the spring not being sufficiently rigid to overcome friction and inertia of the piston and pencil motion; or, from the weakness of the spring, the pencil may be carried to the limit of its motion and held at the stop by a higher pressure than indicated by the scale of the diagram, giving an appearance of flatness of the diagram between the points where the actual pressure is greater than the pressure that can be truly indicated.

Load Factor and Heating of Electrical Equipment—What is the effect of load factor on the heating of generating machinery? R. L.

The effect of load factor on the heating of any piece of electrical apparatus is indefinite, and two machines of equal capacities, operating under the same maximum loads and the same load factor, may show widely different temperatures, depending on the shape of the load curves. For example, a peaked load curve with gentle slopes will give lower temperatures than a flat-topped curve with steep slopes down to minimum loads in which the load remains near the maximum for a considerably longer time, thus causing higher temperatures, even though the average load, and therefore the load factor, may be the same in both cases. Thus it is the shape of the load curve showing the duration of maximum load that determines the heating effect, rather than the load factor which is determined by the ratio of the average to the maximum loads.

Measuring Angle of Advance of Eccentric.—How can measurement be made of the number of degrees of angle of advance at which the eccentric of a slide-valve engine is set? C. M.

Turn the engine wheel in the running direction until the valve rod arrives at the middle of its travel in and out of the stuffing-box gland. This will bring the eccentric at right angles with the engine center line, and the crank angle with the center line will be the angle of advance of the eccentric. With the engine standing in that position, locate a mark on the face of the flywheel to correspond with the end of a stationary pointer that has been brought up to the face of the wheel. Next turn the wheel forward until the crank is at the dead center and locate the point on the face of the flywheel which then corresponds with the stationary pointer, and set off the distance between the points on the edge of a strip of paper laid on the face of the wheel. This distance will be the length of arc of the angle of advance measured on the circumference of the wheel, and the number of degrees will be the distance in inches multiplied by 360 and divided by the circumference of the wheel in inches, taken at that part of the face where the distance was measured.

Trapping Condensate Formed in 8-in. Header—What should be the capacity of a steam trap to discharge the drip from condensation of steam in an uncovered 8-in. steam header 50 ft. long, the steam pressure being 75 lb. gage? G. H.

The rate of condensation should be based upon the lowest probable temperature of surrounding atmosphere. Assuming this lowest temperature to be 50 deg. F., then as the temperature of steam at 75 lb. gage pressure is 320 deg. F., the difference of temperature inside and outside of the pipe would be $320 - 50 = 270$ deg. F. and the rate of heat transmission would be about $270 \times 3 = 810$ B.t.u. per hour per square foot of pipe surface. The external diameter of standard 8-in. pipe is 8.625 in. and, neglecting fittings, the external surface of the header would be $8.625 \times 3.1416 \times 50 \div 12 = 112.9$ sq.ft. and the radiation of heat would be at the rate of $112.9 \times 810 = 91{,}449$ B.t.u. per hour.

The latent heat of steam at the pressure of 75 lb. gage is 893.9 B.t.u. per pound, and the trap would be required to handle as a maximum, $91{,}449 \div 893.9 = 102.3$ lb. of condensate per hour. It should be borne in mind that this estimate of trap capacity applies only to condensate formed in the 8-in. header and does not include drips from other piping and connections or drainage of slugs of water from the boilers that discharge into the header.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications.—Editor.]

Superheaters and Superheat

The following comments on superheater design and performance are excerpts from a paper read by Arthur D. Pratt[1] before the Engineers' Club of Philadelphia. It is understood that a fuller report of this paper will appear in the journal of the club.

A SUPERHEATER should ordinarily be so located as to be swept by all the products of combustion, that is, the same gases that pass over the boiler surfaces should also pass over the superheater. An arrangement by which a portion of the gases are bypassed from the furnace to the superheater direct, without passing over any of the boiler surface, may lead to a decrease in combined boiler and superheater efficiency, even though the gases after passing over the superheater are passed over that portion of the boiler surface beyond it. This decrease in combined efficiency would result from the fact that the mixed gases, after passing the superheater, would be at a somewhat higher temperature than if all gases had passed over the boiler surface before the superheater.

While the location should ordinarily be such that all the surface will be swept by all the gases, it must also be such as to allow easy access for inspection and cleaning. Suitable handholes should be supplied in order that superheater headers and tubes may be easily inspected. Expanded tube joints give the best construction, the tube ends being flared. No flanged joints should be exposed to the hot products of combustion.

Safety-Valve Equipment

All superheaters should be equipped with safety valves, which should be set slightly lower than the boiler valves. With the superheater valves blowing first, a flow of steam through the superheater is assured should the load be suddenly thrown off the boiler, and there will be no danger of burning superheater tubes, as would be the case were the boiler valves to blow first. Again, the use of safety valves on the superheater outlets will prevent highly heated steam passing to the auxiliaries should the main power units be suddenly shut down.

Superheaters should be so arranged that all portions may be thoroughly drained; otherwise in intermittent service slugs of water may be carried over to the prime movers with disastrous results.

In general, whenever possible, it is well to avoid using a saturated steam connection on a superheater boiler. The amount of steam that may be taken through a very small saturated connection is not ordinarily recognized, particularly where there is a slight pressure drop through the superheater, and the amount of steam passing through the saturated opening may be a large proportion of the total steam being generated. This is especially so where a number of boilers are provided with saturated steam outlets and where it is possible to draw more than the proportionate amount of steam from one or more of the boilers. Under such conditions the superheat might become excessive, or if a large enough portion of the steam leaves the boiler through the saturated opening some or all of the superheater tubes would without question be bypassed and become distorted or burned. If saturated connections are necessary, it is well to limit the size of the opening to one that will discharge not more than fifteen per cent of the total steam generated.

The question is frequently raised as to the effect of the installation of a superheater on the efficiency of the boiler in which it is placed. Unquestionably, a properly designed and properly located superheater will tend toward an increase in the combined efficiency of boiler and superheater, and it is common practice in Great Britain to guarantee from 1 to 1½ per cent greater efficiency for a boiler equipped with a superheater over that for a boiler not so equipped. While the higher efficiency would certainly exist, such increase would be extremely hard to prove, owing to the great difficulty in reproducing exactly the same conditions of operation.

For a given boiler capacity measured by the number of B.t.u. delivered in the steam generated the weight of coal which it is necessary to burn will be less for a boiler equipped with a superheater than for a boiler not so equipped by an amount corresponding to the greater efficiency of the superheater units. Too often, however, this same statement is considered to apply to a given weight of steam delivered whether saturated or superheated. Obviously, this is incorrect, since an additional number of heat units must be evolved in the boiler furnace for the superheating of a given weight of steam at a given pressure.

Superheater Performance

The factors affecting superheater performance will be considered under three general sets of conditions, the first of which assumes a given fuel, a given set of combustion conditions and a variation in the capacity or conditions under which the boiler is operated.

It is entirely possible over a fairly wide range of boiler capacity to burn a given fuel under conditions that are within reasonable limits constant: that is, to give reasonably constant weight of products of combustion per pound of fuel burned.

If the combined boiler and superheater efficiency were constant regardless of the boiler capacity, the ratio of gas weight passing over the superheater to steam weight passing through the superheater, as well as the ratio of gas to steam velocity, would also be a constant, regardless of capacity. Under such conditions the superheat would be approximately constant at any boiler rating developed. The problem is not as simple as this, as the efficiency of a boiler will ordinarily fall off with increase in boiler capacity, the extent of such decreased efficiency being dependent upon the fuel and the apparatus that is used for its combustion.

As the boiler capacity increases the weight of steam flowing through the superheater increases almost in direct proportion to the capacity, while the weight of gas passing over the superheater increases at a rate faster than this, such increase in rate, for a given weight of gas per pound of fuel burned, being inversely as the efficiency at the various capacities. This results in an increased heat transfer rate between the gas and the steam. In addition to the increase in transfer rate, there is also an increase in temperature difference between the gas sweeping over the superheater and the steam within the superheater, and the combined increase in the rate of absorption and the increased temperature difference results in an increase in superheat accompanying any increase in boiler capacity. This increase in superheat with increased rating will vary widely, even with a given fuel and a given set of combustion conditions with various designs of furnaces, stokers, boilers and superheaters.

Broadly stated, it follows that the flatter the efficiency-capacity curve the less will be the increase in superheat from a given amount of surface with an increase in boiler capacity, and the greater the falling off of boiler efficiency with increased capacity, the greater will be the increase in superheat.

The superheat at a given capacity will be influenced by the temperature of the feed water. The difference in superheat at a given rating with a boiler run with and without an economizer may be in the neighborhood of 10 per cent. There may also be variations one way or the other of 10 per cent or more if the rate at which the steam is drawn from the boiler suddenly changes, these variations being in the reverse direction to those that come through a gradual change in the rate, the superheat when the steam is drawn

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off suddenly at a higher rate being less than for the periods immediately preceding and following where it is drawn off at a lesser rate.

Variable Combustion Conditions

The second set of conditions affecting the superheater performance assumes a given fuel, a given boiler capacity, but a variation in combustion conditions.

Let us consider first a given coal burned under a boiler of a given size, operating at 125 per cent of its normal rated capacity. If this coal is burned on an efficient stoker, it should be entirely possible to maintain in ordinarily good boiler-room operation 13 per cent CO_2 which would represent some 16 lb. products of combustion per pound of coal burned. If the same coal were burned under the same boiler at the same capacity in a hand-fired furnace, it is probable that under every-day plant operating conditions the average CO_2 would not exceed 10 per cent, which would correspond to approximately 20 lb. products of combustion per pound of coal fired. If the efficiency of the stoker-fired and hand-fired units were the same, the ratio of the gas weight passing over the superheater to the steam weight passing through the superheater, in the case of the hand-fired unit would be some 25 per cent greater than in the case of the stoker-fired unit and the lower efficiency that would be obtained with hand firing would make this ratio still greater. The weight and velocity of the steam passing through the superheater would be the same in either case; the weight of the velocity of gas passing over the superheater would be more than 25 per cent greater in one case than in the other. Obviously, the superheat from a given amount of surface would be appreciably greater in the case of the hand-fired unit than with the stoker-fired furnace.

With certain fuels, other than coal, the possible variation in superheat with combustion conditions may be even greater than that just indicated.

With oil fuel, operating conditions in a well-managed plant probably approach test conditions more closely than with any other fuel, and for this reason the possibility, or rather probability, of variation in combustion conditions is not as great as with other fuels. As an example, however, of the possibility of varying superheat by a change in combustion conditions, an actual occurrence of a number of years ago may be of interest.

A power plant was installed on a bonus or forfeit basis, the bonus or forfeit to be awarded on the basis of a given number of kilowatt-hours delivered at the switchboard per barrel of oil burned. The boilers were sold under a strict guarantee of efficiency to the company making the complete plant installation and to whom the bonus or forfeit was of greatest interest. Not as much was known of variation of superheat with combustion conditions at that time as at present, nor was it fully realized with how little excess air it was possible to burn oil in a properly designed boiler furnace. The superheaters were designed to give 125 deg. of superheat with the boiler developing its normal rated capacity. During the preliminary tests, which were of considerable duration, it was found that the superheat with a minimum amount of excess air was between 95 and 100 deg. On increasing the amount of excess air the superheat was brought up to between 125 and 130 deg. without changing the rate at which the boiler was being operated, or without reducing the boiler and superheater efficiency below that guaranteed. As the lack of the 25 or 30 deg. of superheat would not reduce the overall plant efficiency to anything like the extent of the additional oil burned to secure the guaranteed superheat through admitting excess air, the oil burners were operated with the absolute minimum excess of air in making the tests to determine the bonus or forfeit.

The third set of conditions affecting superheater performance assumes a given rating, combustion conditions as nearly as efficient as possible with different fuels and a variation of fuels, the furnace being suited to the different fuels in each instance.

Assuming that the efficiency of combustion with all fuels is as high as possible under good operating conditions, the class of fuel used has a greater effect on the superheat that a given amount of surface will give at a given rating, than either of the two sets of conditions that have been considered. How much a change of fuel may affect the superheat is not generally appreciated; further, such an effect comes not only with the change of the class of fuel but with a change from one fuel to another of the same class.

In a plant which recently changed from coal on underfeed stokers to oil fuel with mechanical atomizing burners there was a wide variation in superheat at a given rating with the two fuels. A number of superheat readings at various capacities were plotted for both the underfeed and the oil burner results. These plots showed the following results:

	Superheat	
Per Cent of Rating	Coal, Underfeed Stokers	Oil, Mechanical Atomizers
100	75 deg.	48 deg.
150	97 deg.	58 deg.
200	118 deg.	68 deg.
250	135 deg.	78 deg.

The writer is familiar with two plants identical in boilers, stokers, superheaters and load conditions, burning two different kinds of coal which, while they had approximately the same heating value had very different combustion characteristics. When the first plant was started up the superheat was from 30 to 35 deg. higher than the 200 deg. anticipated from the amount of surface provided. When the second plant was put into service the superheat was approximately 30 deg. lower than anticipated, and though every effort was made to bring the superheat up to the desired amount without reducing the efficiency of combustion this was never accomplished with the coal originally burned in the second plant.

Here we have two identical sets of boilers, stokers, load conditions and superheaters, but with the latter giving superheat approximately 60 deg. apart, due to the variation of the combustion characteristics of the two different coals.

It is of interest to note that during the recent period when it was difficult to obtain coal the second plant was forced to change its source of coal supply. When this change was put into effect the superheat instead of being 25 deg. below the anticipated amount was some 25 or 30 deg. higher than that for which the superheater was designed.

Too frequently superheaters are designed and fabricated before it is possible to determine operating conditions. Either the purchaser does not know what fuel, furnace or stoker he will use, or the capacity at which he will operate, or his ideas on the subject change between the time of purchase and the time of installation. In the case of new installations even with fuel, furnace, stoker and capacity known, the superheater builder cannot know the operating conditions and may design his apparatus for what is ordinarily considered average conditions for the class of plant in which the superheater is to be installed. The superheat may then fall below or above the anticipated amount because of operating conditions which may be better or worse than the average.

Still Doing Business

An erroneous reference to the recent dissolution of the Standard Shipbuilding Corporation appeared in the March 22 issue in connection with the biographical sketch of John Hunter. The Standard Shipbuilding Corporation has not been dissolved, but on the contrary, is still actively engaged in building several boats.

Alcohol fuel for automobile engines is coming into use in Pernambuco, Brazil, where gasoline costs $5.83 per case of ten gallons. Alcohol costs but $1.94 for the same quantity, but it causes rusting of the valves. To prevent this, 5 per cent of gasoline or kerosene is added. It is claimed that the power produced and fuel used are about the same for alcohol as for gasoline.

The primary horsepower developed in the city of Los Angeles, Cal., in 1919 amounted to 94,876, or 46.7 per cent more than in 1914.

Development of Electrical Supply in Norway and Sweden

The Swedish government has organized a committee to work out a uniform plan for the electrical supply of the country. This committee has lately reported that at the present time southern Sweden's requirements of electricity can be supplied by utilization of existing water power. The waterfalls of Lagan and Nissan can produce 400,000,000 kw-hr. and the falls in Blekinge can give a further 600,000,000 kw.-hr. The increase in the demand for the last few years has been very marked, and it is expected that after ten years the water-power reserves of southern Sweden will be insufficient and that the country will have to draw on the resources of the northern part, especially the Trollhättan hydro-electric works.

The demand for electric energy in Norway has grown so much during the last few years that most of the larger towns have been forced to ration electric current. This has been the case in Kristiania and Bergen, and these two towns have lately been at work with all energy to increase their power supply. The municipality of Bergen has acquired the Torfis waterfalls and intends to erect a power plant there shortly. The electricity works of Bergen have recently completed a 50,000-volt transformer plant at the power station of Björlo, and a large company, the Bergen Municipal Power Co., has been formed through co-operation of a number of communities for the purpose of utilizing the waterfalls in the vicinity of Bergen. The company has a capital of 9,000,000 kroners; 6,000,000 of this amount have been subscribed by the municipality of Bergen, which has also given it a loan of 20,000,000 kroners.

British Oil-Engine Progress

In a paper on "Oil Engine Progress," by J. L. Chaloner, read before the Feb. 24 meeting of the Diesel Engine Users' Association, the author referred in very favorable terms to the position of the British internal-combustion engine industry and the lines on which it was likely to develop in relation to continental and American competition. He gave figures in support of a statement that of the high-pressure engines in England 25 per cent worked on the mechanical-injection principle and that 50 per cent were two-stroke-cycle engines. In the case of medium-pressure engines 96 per cent worked on the mechanical-injection principle while 58 per cent were two-stroke-cycle engines. These data showed the tendency toward the two-stroke design and the position which the mechanical-injection principle already held with regard to the heavy oil engine. He urged the imperative necessity of further flexibility on the part of the engines in order that heavy-grade oils might be used to a greater extent for internal-combustion engines. The efforts made in England to simplify oil-engine construction have met with greater success than either on the continent or in the United States. The main feature of the high-compression oil engine in England is that it does not use as high a compression as a Diesel engine, and at the same time it dispenses with the air compressor and the necessary air service system.

Various methods of fuel atomization and of air, mechanical and gas-pressure injection were described and compared. Mr. Chaloner considered that the gas-pressure system might be discarded as a practical solution of the fuel-injection method for internal-combustion engines; that the principle involved was only suitable for light fuels and the mechanical difficulties were many; that the greater portion of the charge was in contact with burnt gases, which made it difficult to bring the combustion air into contact with the fuel within the limited time of the combustion period.

The important part which turbulence played in the combustion process had been fully realized and it had been suggested that with mechanical injection a very low degree of turbulence was possible. However, the rate of gasification of the oil could be utilized equally well and owing to the absence of the cooling action of the expanding air the conditions were improved. The rate of vaporization was proportional to the surface of each oil globule and hence in the case of higher atomization the degree of turbulence increased in proportion. The general tendency of introducing long-stroke designs for marine and other slow-running engines produced conditions favorable to the application of mechanical injection.

The author did not suggest that the problem of mechanical injection had been solved, but that the results obtained so far showed that satisfactory progress was being made. The prospects were great and the possibility of a two-stroke-cycle engine without any mechanically controlled valves justified the closest attention by engineers who were anxious to further the progress of the oil engine. In the opinion of the author the two-stroke-cycle mechanical-injection engine would materially assist in bringing the various types of internal combustion into line.

And Here Is Another One

In the *Boston Sunday Herald* of March 6, 1921, appeared an account of the work of Chester S. Hathaway, of Westboro, Mass., who claims to be the originator and perfecter of a power device similar to the "Garabed free energy machine," which was so much talked about two or three years ago. The machine is described as an electrical device for furnishing light, heat and power by which the requirements of a ten-room house may be met at a cost of thirty-five cents a month.

The article describes a typical apparatus consisting of a 110-volt battery and a motor and generator of equal power. Presumably, the battery is to furnish power to run the motor, which is to drive the generator, and the latter in turn is to charge the battery. Most engineers at this point begin to wonder what takes care of the various electrical and mechanical losses of the apparatus. This minor difficulty quickly becomes insignificant beside the really important accomplishments of Mr. Hathaway.

The newspaper article quotes him as follows: "I cannot set up the machinery for I am not an electrician, but after the mechanism has been set up I can do certain things to the generator and to the wires so that it will develop 220 volts as easily as it would develop 110 if it had been run in the normal way." Mr. Hathaway very modestly does not pretend to understand the "scientific significance" of what he does, but, to quote again, "I can do it."

It is then explained that 110 volts of the 220 will keep the battery fully charged, with a surplus of 110 volts for use as may be desired. It seems a pity that someone did not explain to Mr. Hathaway and the newspaper writer the difference between units of electromotive force and units of power.

The last time the device was operated was in 1911, when "it ran for seven months, 24 hours a day, keeping a 20-watt electric light burning, the ordinary automobile battery through which this lamp was supplied being kept charged to the full all the time." Since then Mr. Hathaway has considered it too risky to have the machine in working order because of the efforts of his enemies to steal the secret. These efforts were carried so far that he was beaten to insensibility and left lying by the roadside at night in an attempt to steal the plans of the apparatus.

The entire account is so manifestly absurd to anyone with the most elementary knowledge of engineering principles that it might be treated as a joke or else beneath notice. Probably, however, it was read by many people who could not understand its absurdity and who only saw an account of a wonderful new power device. In justice it should be said that no mention was made of any stock being offered for sale, but the story as printed would make an excellent "opening gun" in a campaign to "let the public in on the ground floor."

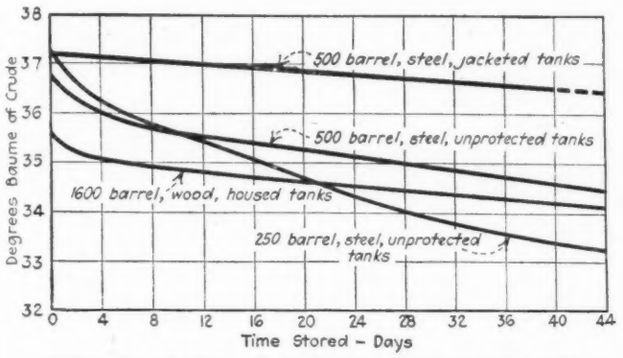
Why will a reputable newspaper print articles on technical subjects without first submitting them to competent technical reviewers?

A fire-resisting plastic material consists of equal weights of paper paste and a mixture of equal parts of calcium carbonate and magnesium silicate.

Evaporation Loss of Crude Oil in Storage*

BY J. H. WIGGINS†

Crude oil, if exposed to the air, will continue to evaporate for a long time. If the mixture of air and vapor at the surface of the oil is nearly pure air, such as is the case when wind blows across it freely, the rate of evaporation will be most rapid. As a vapor-air mixture above the oil receives a larger and larger percentage of vapor the rate of evaporation decreases. If the mixture finally contains as much of the vapor as it can possibly hold under given temperature and pressure conditions (saturation point) then evaporation will cease. In the case of a tank that has a gastight cover the oil will evaporate until the air under the roof is saturated. If this mixture of air and vapor is held with no entrance of fresh air evaporation will cease. But in this closed tank there is a certain amount of expansion of the gas under the roof owing to its change in temperature, which causes the tank to "breathe." This leads to the conclusion that the temperature of the gas under the roof should be kept as nearly constant as possible. Combined then, the effect to be attained to practically eliminate evaporation is to confine the space above the oil in a tank and to keep that space at a constant temperature. Then the only evaporation loss will be that necessary to saturate the space above the oil.



SHOWING EFFECT OF EVAPORATION ON GRAVITY

To accomplish this tank roofs should be absolutely gastight with automatic pressure valves. In some cases they should also be housed, or both housed and water-sealed on top in addition.

There are a number of effects resulting from evaporation of crude oil. In the first place the gravity of the crude becomes heavier, caused by the escape of the lighter fractions of the mixture. Second, as evaporation continues the gravity of the fraction that is actually going through the process of evaporation becomes heavier and heavier. Third, in conjunction with the heavier gravity of the fraction evaporation, the boiling point of this fraction rises. Fourth, the average gravity of the whole portion of the crude lost by evaporation becomes heavier.

The table shows the classification for presenting data concerning the aforementioned effects of evaporation. A study of the difference in effect between one kind of tank and another kind will give the relative values of each type toward the elimination of evaporation.

As oil evaporates its Baumé gravity becomes heavier owing to the escape of the lighter fractions which are part of it. The effect on several oils from this viewpoint under various conditions of storage is shown in the figure. It is seen here that the 250-bbl. steel unprotected tank gives the greatest loss in Baumé gravity on the oil stored in it, whereas the 500-bbl. steel-jacketed tank shows the least loss in this respect.

It may be said that with a crude giving a distillate of 84-deg. Baumé all of this degree Baumé will evaporate, while very little of the 80-deg. fraction distilled next will have evaporated and even less of the still lower fractions. In other words, the practical result is that if a crude contains 1 per cent of gasoline having an average gravity of 84 deg. (practically all of this may be composed of pure hydrocarbons of from 80 to 88 deg.), the first 1 per cent evaporated will all be this 84-deg. fraction. After this has disappeared then the 1 per cent fraction distilled immediately below it, which may be 80 deg., will start evaporating. This empirical rule is not exact, but it is the interpretation of the results of over 200 distillations before and after evaporation, and it checks with actual field results.

EVAPORATION OF CRUDE OIL

Kind of Tanks Covered*	Season	Gravity of Oil, Deg. Bé.	Effect Shown on
1	Summer	35 to 37	Baumé gravity of crude.
2	Summer	35 to 37	Baumé gravity of fraction evaporating.
3	Summer	35 to 37	Boiling point of fraction evaporating.
4	Summer	35 to 37	Baumé gravity of portion lost.

* 1. 250-barrel, steel unprotected tanks. 2. 500-barrel, steel unprotected tanks. 3. 500-barrel, steel jacketed tanks. 4. 1,500-barrel, wood housed tanks.

*Extract from Bureau of Mines Bulletin No. 200.
†Petroleum Engineer, U. S. Bureau of Mines.

Utilization of Heat Content of Hot Springs as a Source of Power*

The sensible heat contained in natural springs of mineral water is often considerable, and when the flow of the spring is sufficiently large, it constitutes a source of energy that is worthy of notice. For instance, by using this heat for the production of steam to be used in a low-pressure turbine, the different springs of Aix-les-Thermes, in France, would supply about 100 hp., those of Bourbon-Lancy, 600 hp., and those of Hammam-Meskoutine, in Algeria, 12,200 hp., or sufficient power to operate a large section of the Algerian railways.

The writer proposes to pass the water from the springs through one or more "auto-evaporators" in series, the device being a closed vessel from which the air has been exhausted and in which the water trickles over suitable evaporating surfaces. Steam from this vessel is fed to the first stage of a special low-pressure turbine. The water which reaches the bottom of the first evaporator without being vaporized passes into a second evaporator which is connected to the second stage of the turbine, and is therefore at a lower pressure; in this vessel some of it is vaporized and fed to the turbine, and the remainder passes into a third evaporator, connected to a third stage of the turbine, and so on, until no more heat can be economically removed from the water. The last stage of the turbine exhausts into a condenser of either the jet or surface type. To dry the steam and increase the temperature range and consequently the cycle efficiency, the steam from each successive evaporator before entering the turbine is passed through a superheater, heated by a coal furnace and its temperature increased to about 70 deg. F. of superheat. The quantity of heat necessary to accomplish this will be very small, and will add little to the operating expenses. The multi-stage turbine may be replaced by several single-stage turbines for different pressures, but the former is claimed to be more efficient. The different stages of the turbine must be designed to handle both the steam exhausted by the preceding stages and that from its own evaporator.

Fourteen years ago, or, to be exact, on April 16 and 17, 1907, the Engineering Societies Building was dedicated, the exercises including addresses by Andrew Carnegie, the donor; Charles Wallace Hunt, past president of the American Society of Mechanical Engineers; Arthur C. Hadley, president of Yale University, and others of international prominence, the dedicatory prayer being offered by the Rev. Edward Everett Hale. All the original holding societies had previously moved into the building, and Frederick P. Fish, of the American Institute of Electrical Engineers, made the first professional use of its facilities on Jan. 8 when he addressed a meeting of the American Society of Mechanical Engineers on "The Ethics of Trade Secrets."

*Abstract of article by P. Caufourier in "Le Genie Civil," Jan. 8, 1921, p. 37.

Washington News

By Paul Wooton
Washington Correspondent

Officials in Washington are alarmed at the slump in coal production. For many weeks production has been millions of tons below the normal maximum output. Forty per cent of the coal-carrying equipment of the railroads is idle. Coal is being bought on a hand-to-mouth basis, evidently with the idea that lower prices are coming. An exhaustive statement reviewing the situation has been issued by the United States Geological Survey. A high Government official, who is particularly well informed as to the coal situation, points out that the coal industry is on a very unstable basis and that a grave risk is being taken by allowing coal-carrying equipment to remain idle. He fears that the minute prices turn upward, there will be a rush of buying that will carry quotations quickly into unreasonable figures. There was an example of such a tendency when the announcement was made of the failure of operators and miners in Great Britain to reach an agreement. The advent of a strike by the British miners will have a very decided influence on prices in this country, it is believed, despite the fact that our exports are limited by coal-handling facilities at ports. In that connection, however, attention is called to the fact that the capacity for handling export coal is increasing rapidly.

The effect of a strike in Great Britain would make its influence felt here more in a psychological than in a material way. As was the case last fall, the very fact that foreigners were scrambling for our coal did much to add to the volume of the buyers' panic in which consumers bid against themselves until prices reached prohibitive levels.

Priority orders on coal proved so unpopular that it is doubtful if they will be resorted to again. The Interstate Commerce Commission has made it clear that large consumers who fail to protect themselves by contract contribute importantly to a coal situation such as existed during the latter half of 1920. It would not be surprising if the Interstate Commerce Commission were to point out to applicants for priorities next fall that relief could not be extended, since they had neglected to secure fuel supplies during the several months that coal was available. The road-building program during the coming summer will call for 25 per cent more cars than have ever been needed before to handle road materials. The housing situation is such that large amounts of building material must be transported. In addition, sight must not be lost of the danger of a major transportation disturbance as an outgrowth of the labor policy being adopted by the carriers.

While officials are inclined to blame consumers for not making contracts, they feel also that the railroads could do more toward encouraging coal transportation by authorizing reductions in rates and that the operators could stimulate buying if they were willing to make price concessions during a period such as has existed since the first of the year.

Refrigerating Engineers Organizing in the Northern States

Since the annual meeting in December the National Association of Practical Refrigerating Engineers has been active in the organization of new subordinates, plans for five or six now being under way. Immediately after the convention a subordinate was organized in Philadelphia, which will be the headquarters for the next annual convention, and there are prospects for others in the East. In the Middle West, a subordinate was organized in Indianapolis on Feb. 25. There were 21 charter members and Owen J. Cox, chief engineer of the Indiana Refrigerating Co., was elected president. In Milwaukee on March 21 a subordinate was formed with 28 members and John M. Jung, master mechanic of the Blatz Brewing Co., as president. In this locality refrigerating engineers are numerous and there are prospects for a good live organization.

In addition to this organizing activity the national body has been busy along educational lines. A new course in refrigeration has been practically completed. The aim is a nation-wide organization that will be representative of practical refrigerating engineers throughout the country, and in the last few months an excellent start has been made.

New Publications

ACCEPTANCES, TRADE AND BANKERS'. By Park Mathewson. Published by D. Appleton & Co., New York City, 1921. Cloth; 5 x 7¼ in.; 372 pages; 42 illustrations.

The book describes the theory, procedure and practice of acceptance payment, gives condensed rulings of the Federal Reserve Board on acceptance procedure and shows the forms used down to 1920. It also outlines the plan of a campaign to induce the use of trade acceptances. It is not written only for bankers, but is directed also to manufacturers, jobbers, merchants, publicity managers, salesmen and business men in general.

COMPRESSED AIR. By Theodore Simons, Professor of Mining Engineering, University of Montana. Second Edition. Published by McGraw-Hill Book Co., Inc., New York. Size 6 x 9 in. 173 pages; 42 illustrations. Price, $2.

This treatise on compressed air, being in the nature of a textbook, deals with the basic natural laws and principles underlying the production and use of compressed air, and includes short descriptions of various types of air compressors. The first three chapters dealing with the law of gases and air are particularly lucid. The method of treatment is such that the reader is allowed to understand each stage in the development of the several laws of gases. Other subjects dealt with are compound compression, volumetric efficiency compression at altitudes, indicator diagram, cooling water, transmission and use of compressed air and description of air compressors. In the discussion on compressors the author fails to deal with many of the new designs of high-speed machines. The volume is in all respects a valuable treatise on the subject.

ELEMENTS OF FUEL OIL AND STEAM ENGINEERING. By Robert Sibley, B.S., and C. H. Delany, M.E.E. Second Edition: Published by McGraw-Hill Book Co., New York; Size 6 x 9-in., 443 pages; 248 illustrations. Price, $5.

This volume is a treatise on the use of oil fuel in power plants, especially central stations. While the major part of the contents is devoted to this phase of engineering, a considerable portion is unfortunately given over to discussion on thermodynamics and kindred subjects. These subjects are broad enough to be dealt with in an exclusive work. The volume on the whole is well written and covers the field of oil fuel in a very comprehensive manner. One very commendable feature is the inclusion of Fire Underwriters' Requirements on Oil Storage. For those who are investigating the possibilities of oil fuel the description of equipment, operation, etc., will have a ready appeal. On the question of relative coal and oil fuel efficiencies the authors have laid too much stress on the more favorable oil efficiencies. It is becoming fairly well appreciated that there is no great difference in furnace efficiencies with the two fuels. Likewise, contrary to the authors' statement, a well equipped stoker coal-burning plant will not require any considerable amount of labor over the labor in the oil-fired plant. The volume should be of decided interest to engineers and increase the fund of available information on the use of oil as fuel.

Obituary

Walter F. Deming, president of the Deming Co., Salem Ohio, died of heart failure March 19 at his home in Salem. He was prominent in civic affairs and spent the greater part of his business life in the manufacture of pumps. He was born in Salem in 1853, was graduated from Salem High School in 1872, and a few years later went into business with the Silver and Deming Manufacturing Co., becoming secretary and general manager. This company took up pump manufacturing in 1880 and ten years later a division was made by which the Deming Co. was organized to take over the pump business. Mr. Deming was treasurer of the new company and in 1894 succeeded his father as president, retaining the position up to the time of his death.

Personals

A. V. S. Lindsley has been appointed sales manager of the Connecticut Light and Power Co., Waterbury, Conn.

Joseph A. Polson has been appointed associate professor of steam engineering at the College of Engineering of the University of Illinois.

George Sutherland has left Stone & Webster, Inc., Boston, to accept a position with the United Electric Light and Power Co., New York.

A. A. King has taken the position of chief electrician with the Phelps-Dodge Corp., Copper Queen Branch, Mines Department, Bisbee, Ariz.

Wallace B. Kirke, previously with the General Electric Co., has been appointed assistant to the chief electrical engineer of the New York Edison Co.

Nathaniel Greene Herreshoff, naval architect and engineer, was elected to honorary membership in the A. S. M. E. at the council meeting at Syracuse.

Louis F. Klauder has given up his work as construction engineer for the Philadelphia Rapid Transit Co., to enter the consulting engineering field. He will specialize on plant design and construction.

Fred A. Scheffler, general manager of the Fuller Engineering Co., has been chosen president of the Edison Pioneers. Mr. Scheffler joined forces with Edison in 1881 and designed for him an experimental electric locomotive.

J. R. Bibbins has been appointed head of the newly established department of transportation and communication of the United States Chamber of Commerce. Mr. Bibbins has had considerable experience as a consulting engineer in the United States and Canada.

D. A. Hegarty, formerly with the Western Light and Power Company of Boulder, Colo., has been appointed manager of the Brush Electric Company at Galveston, Texas, succeeding W. R. Phipps. Both these plants are owned by the Henry L. Doherty interests of New York, and Mr. Phipps has been transferred to the New York office of the company.

Haylett O'Neill, occasional contributor to *Power*, and well-known engineer, has associated himself with William D. Leeds, 29 Broadway, New York City, Sales Agent for Langdon Coal Co., Smooth Hill Coal Co. and Thomas J. Lee Collieries, and is prepared to furnish counsel and advice with regard to the selection and burning of fuel, as well as to take orders therefor.

Society Affairs

The American Association of Engineers will hold its seventh annual convention in Buffalo May 9-11.

Hartford Branch, A. S. M. E., will meet April 11 to hear A. W. Honywill, Jr., of the Terry Turbine Co., speak on "The Corrosion of Metal Surfaces."

San Francisco Section, A. S. M. E., will hold a meeting April 14 at which the gas-fired steam plant of the San Joaquin Light and Power Co. will be described by A. Y. Meidell, representing the company.

The Providence and Textile Sections of the A. S. M. E. will hold a meeting April 12 in the afternoon and evening. One of the papers will be "The Effect of the Variable Speed Motor on the Heat Balance in the Finishing Plant," by Warren D. Lewis.

The Detroit Engineering Society plans to meet April 14 to consider "Industrial Electrical Heating." C. F. Hirschfeld, of the Detroit Edison Co., will trace the development in the use of electric power up to the present time and will describe industrial heating processes in which electricty can compete with fuel.

Miscellaneous News

A Power Test Code for reciprocating steam engines will be presented at the A. S. M. E. Spring meeting. A test code for steam turbines is also being prepared.

The Bituminous Coal Industry is one of ten now being analyzed by the committee on elimination of waste in industry of the Federated American Engineering Societies. L. W. Wallace was appointed chairman of the committee and was later chosen as secretary of the Federation. The work being done under his leadership is intended to disclose the causes of whatever waste there is in ten leading industries. The report is expected to be ready April 1.

Business Items

The Hooven, Owens, Rentschler Co., maker of Hamilton engines and machinery, announces that Mr. F. H. Boyer, Hubbell Building, Des Moines, Iowa, has been appointed one of the agents in the Des Moines district.

Trade Catalogs

The Manistee Iron Works Co., Manistee, Mich., is distributing a little folder containing some interesting pictures and diagrams of "Roturbo" pumps and condensers.

The Anderson Foundry and Machine Co., Anderson, Ind., has issued a new 8 x 11-in. 32-page catalog of 30- to 300-hp. solid injection oil engines of the two-stroke-cycle, vertical, stationary type.

The Fuller-Lehigh Co., Fullerton, Pa., is distributing an 8 x 10½-in. 28-page bulletin on the use of pulverized coal under steam boilers. The company's products include burners and all the necessary crushing and conveying apparatus for a powdered-coal burning plant.

Busch-Sulzer Bros. Diesel Engine Co., St. Louis, Mo., is sending out a 12-page pamphlet bringing out the use and advantages of Diesel engines in large hydro-electric and auxiliary plants. Installations all over the world above 2,000 b.hp. are cited, the general dimensions of the Busch-Sulzer four- and six-cylinder type C engines are tabulated, and an interesting letter from the Sanitary District of Chicago is presented showing why the Diesel engine was given preference over steam equipment in the new Calumet power plant.

FUEL PRICES

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine-run basis, f.o.b. mines):

Coal	Market Quoted	Mar. 22, 1921	Mar. 29, 1921
Pocahontas,	Columbus	$3.25	$3.25
Clearfield,	Boston	2.50	2.00@3.00
Somerset,	Boston	3.00	2.40@3.60
Pittsburgh,	Pittsburgh	2.25	2.00@2.50
Kanawha,	Columbus	2.25	2.35
Hocking,	Columbus	2.15	2.00@2.50
Pittsburgh No. 8	Cleveland	2.15	2.20@2.35
Franklin, Ill.,	Chicago	3.35	3.25@3.50
Central, Ill.,	Chicago	2.00	2.00@2.50
Ind. 4th vein,	Chicago	2.65	2.25@3.00
Standard,	St. Louis	1.85	1.75@2.00
West Ky.,	Louisville	2.50	2.50
Big Seam,	Birmingham	2.95	2.70@3.00
S. E. Ky.,	Louisville	2.75	2.50@2.85

DIESEL FUEL OIL

New York—On March 23, Port Arthur light oil, 23@25 deg. Baumé; 30@35 deg., 7c. per gal., f.o.b. Bayonne, N. J.

Chicago—24@28 deg. Baumé, $1.10@$1.20 per bbl.; 32@34 deg., 3¾@4¼c. per gal.

Cincinnati—In 8,000-gal. tank cars; 26@28 deg., 9½c. per gal., 30@34 deg., 10c.

St. Louis—March 26 prices f.o.b. cars, tank lots; 24@30 deg. Baumé, 2.1c. per gal. 32@34 deg., 3c.; 16@20 deg. 3¾c. per gal.

Baltimore—On March 28, f.o.b. Baltimore in tank-car lots: 26@28 deg. Baumé, 6¼c.; 30@34 deg., 4¾c., 14 deg., 4¾c. or $2 per bbl.; 16@20 deg., 4¼c. or $1.90 per bbl.

Pittsburgh—On March 28, f.o.b. refinery: Pennsylvania, 36@40 deg., 4½@5c. per gal.; Oklahoma, 22@24 deg., 60@70c. per bbl., 24-26 deg., 70@80c., 26@34 deg., 80@90c.; gas oil, 32@34 deg., 3c. per gal., 36@38 deg., 3¾c., 38@40 deg., 4c.

New Construction

PROPOSED WORK

Me., Portland—J. H. Feeney, 30 State St., is in the market for laundry equipment including 50-70 hp. boiler, 20-30 hp. engine, etc.

Mass., North Dighton—The Mount Hope Finishing Co. will soon award the contract for a 1 and 4 story textile manufacturing and storage building on Spring St. About $250,000. H. M. Burke, Engr.

Mass. Quincy Adams (Boston P. O.)—L. Grossman & Sons is in the market for 2 electric driven coal loaders, also other equipment for handling coal.

Mass., Russell—H. T. Boynton is in the market for one 5 hp., single phase, 60 cycle, 110 volt motor.

Mass., Worcester—R. H. Long Co., Framingham, is having plans prepared for a 6 story, 75 x 600 ft. main building, 6 story, 75 x 300 ft. factory and a 6 story, 75 x 100 ft. factory on Millbrook St. along the tracks of the Baltimore & Maryland R.R. About $1,500,000. F. H. Clark, Engr.

Conn., West Hartford—The School Comn. will soon award the contract for a 2 story, 62 x 200 ft. junior high school with a 60 x 80 ft. addition on Whiting Lane. A steam heating system will be installed in same. W. T. Marchant, 36 Pearl St., Hartford, Archt.

N. Y., Buffalo—The Saturn Club, Delaware Ave., is having plans prepared for a 3 story, 150 x 200 ft. club house. About $400,000. Bley & Lyman, 232 Delaware Ave., Archts.

N. Y., Mt. Vernon—The Willow Brook Dairy Co., 58 Fulton Ave., is having plans prepared for a dairy plant including a power plant, etc. About $250,000. McCormick Co., Century Bldg., Pittsburgh, Pa., Archt.

N. Y., New York (Manhattan Boro.)—The Sallye Realty Co., 358 6th Ave., plans to build a 12 story office building at 149-151 West 23rd St. About $500,000. L. Frazen, Pres.

N. Y., New York (Manhattan P. O.)—S. Wachatinsky, 308 East Bway., plans to build a 3 story, 148 x 250 ft. garage including a steam heating system on Broome and Tompkins Sts. About $350,000.

N. Y., New York (Manhattan Boro.)—W. Whitehill, Archt. and Engr., 12 Elm St., will receive bids about April 15 for a sub-station on Greene St., for the New York Edison Co., 130 East 15th St. About $160,000.

N. Y., Silver Creek—The Bd. Educ. will receive bids until April 19 for a 3 story, 157 x 170 ft. high school. About $300,000. Harvey S. Horton, Prudential Bldg., Buffalo, Archt.

N. Y., Wende (Millgrove P. O.)—The Bd. of Supervisors, City and County Hall, Buffalo, will receive bids until April 12 for a 1 story, 44 x 396 ft. penitentiary and a 1 story, 90 x 92 ft. boiler house, etc. About $450,000. G. E. Speich, 575 Ellicott Sq. Bldg., Buffalo, Archt. G. C. Diehl, 575 Ellicott Sq. Bldg., Engr.

N. J., Bridgeton—Custer & Gill, Archts., 328 Market St., Camden, will soon receive sub-bids for a 2 story, 40 x 120 ft. clothing factory here on South Laurel St., for Morvay & Sons,. Motors for small sewing machines will be installed in same. About $25,000.

N. J., Plainfield—The New Jersey State Insane Asylum, Morris Plains, received low bid for installing a heating system in the proposed 4 story, 64 x 147 ft. psychiatric building, to G. J. Tobin, 187 North Ave., $33,449. Noted March 9.

Pa., Altoona—E. G. Burket, 816 11th Ave., plans to build a 4 story, 80 x 146 ft. theatre and office building on 11th Ave. About $300,000. Architect not yet selected.

Pa., Butler—The Butler Memorial Hospital, c/o Boring & Tilton, Archts. and Engrs., 52 Vanderbilt Ave., New York City, will soon receive bids for a 3 story hospital on Brady St., here. About $500,000.

Pa., Hughesville—J. H. and C. K. Eagle, Inc., Franklin and Independence Sts., Shamokin, have purchased a site along the tracks of the Williamsport and North Branch R.R. and plans to construct a 1 story, 58 x 175 ft. silk mill plant and power house. About $50,000.

Pa., Johnstown—The Loyal Order of Moose, Johnstown Lodge No. 48, Bedford St., plans to construct an 8 to 12 story, 50 x 100 ft. lodge building including offices and apartments. About $400,000. H. A. Hamer, Secy.

Pa., Pittsburgh—The Underwriters Bldg. Co., 330 4th Ave., plans to build an 18 story, 70 x 110 ft. office building on 4th Ave. About $1,500,000. Kennedy, Ellington & Associates, 230 5th Ave., Archts.

Md., Baltimore—The Circle Theatre Co., 532 North Calvert St., plans to build a theatre at Park Circle and Reisterstown Rd. About $650,000. E. G. Blanke, Archt.

Md., Baltimore—Dietrich Bros., Pleasant and Davis Sts., will soon award the con-

tract for a 1 story, 190 x 330 ft. fabricating shop on Waverly St. About $350,000.

W. Va., Bluefield—The Bluefield Rotary Club is having preliminary plans prepared for an 8 story hotel. About $500,000. A. Mahood, Archt.

W. Va., Ellenboro—E. J. White is in the market for one 20 hp. boiler, water pumps, one 35 hp. gas engine and one 15 hp. gas engine.

N. C., Raleigh—The State Dept. of Agriculture plans to build a 5 story agricultural building on Edenton St. About $500,000. W. A. Graham, Comr. of Agriculture.

N. C., Raleigh—The State Legislature has appropriated $775,000 for the construction and equipment of various buildings at the North Carolina Central Hospital for the Insane here. A. Anderson, Supt.

N. C., Wilmington—The Carter's Production Wks., 210 South Water St., O. Carter, Mgr., is in the market for a 6 x 8 D. D. D. C. hoisting engine with boiler.

S. C., Columbia—The Columbia Physicians plan to build a 9 story building. Cost between $500,000 and $750,000. Address C. C. Wilson, Palmetto Bldg., Archt.

Fla., Key West—C. Aubuchon and Associates have been granted a franchise to install wells and pumping equipment here to deliver water to the citizens.

Fla., Miami—The County School Bd. voted $300,000 bonds to build 2 story, 100 x 150 ft. schools.

Fla., Miami—The Miami Water Co., 12th and Court Sts., plans to build new water mains and install additional pumping equipment. About $200,000.

Fla., Orlando—Mayor Duckworth engaged Alvord & Burdick, Engrs., 8 South Dearborn St., Chicago, Ill., to survey present lighting plant and prepare estimates for a new plant to be built by city.

Tenn., Memphis—The Tri-State Hotel Co. plans to build a 16 story, 102 x 160 ft. hotel. About $1,500,000. Barnett, Haynes & Barnett, Century Bldg., St. Louis, Mo., Archts.

O., Canton—A. H. Abrams Co., Tuscarawas St., East, plans to build a 3 story, 33 x 200 ft. theatre building on Market Ave., North. About $300,000. H. M. Miller, Archt.

O., Cincinnati—M. F. Quill, Genl. Contractor, 3337 Reading Rd., is in the market for a boiler and engine.

O., Cleveland—J. L. Free and A. W. Henn, 1204 Prospect Ave., have purchased a site at 1227-41 Prospect Ave. and plans to build a 10 story office building on same. About $750,000. Watson Eng. Co., Hippodrome Bldg., Archt. and Engr.

O., Dayton—The Jefferson Main Arcade Co., 908 U. B. Bldg., plans to build a 5 story, 90 x 200 ft. hotel and office building on East 5th St. About $300,000. Architect not yet selected.

O., Dayton—The Rike Kumler Co., 2nd and Main St., plans to build an 8 story, 55 x 190 ft. department store addition on North Main St. About $500,000. Architect not yet selected.

O., New Philadelphia—The Delphia Hotel Co. plans to build a 5 story, 66 x 152 ft. hotel on North Bway. About $300,000. H. B. Frank, Akron, O., Archt.

O., St. Bernard—The city plans an election to vote on $110,000 bonds to improve waterworks and electric light plant.

Mich., Detroit—The Bd. of Fire Comrs., West Larned St., received bids for a 2 story, 80 x 90 ft. pumping station including pumps, air compressors, etc., on the foot of Randolph St., from the Wolverine Constr. Co., 400 Penobscot Bldg., $61,552, and Everett Winters, Detroit Savings Bank Bldg., $63,101.

Mich., Detroit—The city had plans prepared for a group of hospital buildings including an office building, main building with double Y wing on each end, also a 5 story surgical building also a power plant, on Hamilton Blvd. Heating equipment will be installed in same. About $3,000,000. A. Kahn, 1000 Marquette Bldg., Archt.

Mich., Lansing—The Michigan Screw Co. is in the market for miscellaneous machine equipment motor driven also including motor generator set.

Ill., Chicago—Louis C. Bouchard & Co., Archt., 30 North Dearborn St., will soon award the contract for a 4 story, 150 x 150 ft. apartment house including a steam heating system on Lakewood and Loyola Sts. About $350,000. Owner's name withheld.

Ill., East St. Louis—The Dependable Truck & Tractor Co., 231 Collinsville Ave., is having preliminary plans prepared for a 1 story, 100 x 250 ft. truck plant and power house at Washington Park. About $250,000. C. B. Morse, Pres. Reister & Rubach, Murphy Bldg., Archts.

Wis., Kaukauna—The Bd. Educ. is having plans prepared for a 2 story, 60 x 210 ft. senior high school including a steam heating system on Main St. About $250,000. F. A. Towsley, Secy. Parkinson & Dockendorff, LaCrosse, Archts.

Wis., Menominee—The Litnum Bronze Co. is in the market for a belted compressor of 70 cu.ft. replacement per min., complete with unloader and high pressure tank, 30 in. diameter x 6 ft. long, 200 lb. working pressure.

Ia., Dubuque—J. W. Royer, Archt., Urbana, Ill., will receive bids until April 15 for a 3 story school here for the Bd. Educ. Estimated cost, $400,000.

Ia., Grinnell—The Bd. Educ. will receive bids until April 14 for a 3 story school. About $275,000. Proudfoot, Bird & Rawson, 810 Hubbell Bldg., Des Moines, Archts.

Ia., Sioux City—The Bd. Educ., City Bldg., will receive bids until April 18 for a 2 story, 80 x 130 ft. junior high school including a steam heating system on 25th and Nebraska Sts., North. About $325,000. Beauttler & Arnold, 235 Grain Exch. Bldg., Archts.

Minn., Minneapolis—The Hamm Brewing Co., 681 East Minnehha St., St. Paul, plans to build a 12 story, 32 x 165 ft. office building including a steam heating system on 2nd Ave. South and 7th St. About $1,500,000.

Minn., Minneapolis—The McMichael Investment Co., 402 Wilmac Bldg., plans to build a 6 story, 31 x 122 ft. office and store building including a steam heating system at 715 Hennepin Ave. About $250,000.

Minn., South St. Paul—The Bd. Educ. voted $350,000 bonds to build a 2 story, U-shaped high school including a steam heating system. J. R. Stevenson, Clk. Toltz, King & Day, 1410 Pioneer Bldg., St. Paul, Archts.

Minn., Virginia—R. J. McGhee, Secy. of the Bd. Educ., will receive bids until May 1 for a 2 story, 70 x 275 ft. grade school including a steam and mechanical ventilation system, in the North Side. About $325,000. Kelly & Shefchik, 507 Lansdale Bldg., Duluth, Archts. and Engrs.

Kan., Osborne—The city voted April 5 to issue $75,000 bonds to build a power plant. Archer & Stevens, 612 New England Bldg., Engrs.

S. D., Elk Point—The city will soon award the contract for a sewage system including a pumping plant. About $35,000. T. H. Johnson, 319 Iowa Bldg., Sioux City, Ia., Engr. Noted Jan. 4.

Okla., Henryetta—B. Morgan is having plans prepared for a 7 story, 140 x 140 ft. hotel. About $350,000. C. Boller & Bros., 508 Ridge Bldg., Kansas City, Mo., Archts.

Okla., Sayre—The city is having plans prepared for a waterworks system including a pumping station. W. E. Symonds, Clk. V. V. Long & Co., 1300 Colcord Bldg., Oklahoma City, Engr.

Cal., Los Angeles—H. L. Lumbiner, c/o E. Bergstrom, Archt., 1129 Citizens Bank Bldg., plans to build a 12 story office building on 8th St. and Bway. About $750,000.

Ont., Dunnville—E. H. Darling, Engr., Home Bank Bldg., Hamilton, has been engaged to prepare estimates on a filtration plant and pumphouse for this town. About $50,000.

Ont., Gravenhurst—C. S. Cobb, Archt., 71 Bay St., Toronto, will receive bids in April for a 3 story hospital including a vacuum steam heating system, for the Muskoka Free Hospital, 223 College St., Toronto. About $350,000.

Ont., London—The Service Motor Truck Co., Wabash, Ind., will receive bids until April 30 for an auto factory including three 1 story, 110 x 300 ft. buildings and power house on Chelsea Green. About $300,000.

Ont., Saulte Ste. Marie—R. G. Campbell, City Clk., is in the market for boilers and equipment for the proposed 3 story, 90 x 106 ft. court house.

Ont., Saulte Ste. Marie—Finlay & Fowles, Archts., 527 Queen St., East, will soon award the contract for installing a heating and ventilating system in the proposed 3 story, 49 x 140 ft. addition to the high school.

CONTRACTS AWARDED

N. Y., New York (Manhattan Boro.)—The Bensault Constr. Co., c/o C. Kreymberg, Archt. and Engr., 2534 Marion Ave., will build a 5 story, 70 x 100 ft apartment house including a steam heating system on University Ave. About $275,000. Work will be done by day labor.

N. J., Newark—J. M. Brennan, Morris & Shessings, Broad St., here, and J. Leblang, 1482 Bway., New York City, will build a theatre, store and office building including a steam heating system on Summit and Newark Aves. About $500,000. Work will be done by day labor.

N. J., West New York (Weehawken P. O.)—S. M. Adelman, c/o Hyman Rosensohn, Archt. and Engr., 188 Market St., Newark, will build two 5 story, 100 x 125 ft. apartment houses here, including a steam heating system on 19th St. and Hudson Blvd. About $350,000 each. Work will be done by day labor.

O., Cleveland—The Bd. Educ., East 6th St. and Rockwell Ave., has awarded the contract for the installation of a heating system in the proposed 3 story, 220 x 330 ft. high school at Woodland Hills Park, to the Feldman Bros., 10305 Barrett Ave., at $129,500 and in the proposed 3 story, 220 x 330 ft. high school on Hopkins Ave. and East 118th St., to W. B. Van Sickle, 707 Frankfort St., at $134,989. Noted March 29.

Ind., Ft. Wayne—The First and Hamilton Natl. Banks, Berry Sts., has awarded the contract for a 14 story, 100 x 150 ft. bank and office building, to G. A. Fuller Co., 175 5th Ave., New York City. About $1,500,000.

Mich., Detroit—The Bd. Educ., 50 Bway., has awarded the contract for a 2 story, 250 x 405 ft. school including a steam heating system on Seneca and Sylvester Sts., to Bryant & Detwiler, 2336 Dime Bank Bldg., at $1,073,736 and for the Codd School including a steam heating system on Wilson and Blain Aves., to A. W. Kutsche Co., 623 Ford Bldg., at $950,800.

Mich., Detroit—A. Harvey's Sons Mfg. Co., 1st and Woodbridge Sts., has awarded the contract for a 6 story, 85 x 105 ft. warehouse including a steam heating system, conveyors, etc., to Walbridge & Aldinger, 2356 Penobscot Bldg. About $200,000.

Ia., Charles City—The city has awarded the contract for a sewage disposal plant including centrifugal pump and motor, to Ward & Weighton, 516 Davidson Bldg., Sioux City, at $128,331.

Minn., Keewatin—The Water, Light & Power Bldg. Comn. has awarded the contract for one electric motor driven vacuum pump of the Nash-Jennings type, to be of simple size, 220 volt, 3 phase, 60 cycle and one pump, electric motor driven, plunger type, deep well, having capacity of 225 gal. per min. under total head of 250 ft., to J. G. Robertson, 2542 University Ave., St. Paul. About $9,000.

Kan., Caney—The city has awarded the contract for a waterworks system to J. W. Davis, Eureka, Kan., at $27,066 and for engine and pumps, to the Challenge Co., 1404 West 12th St., Kansas City, Mo., at $2,125.

Wyo., Riverton—The Riverton Hide, Wool & Fur Co. has awarded the contract for grain elevator and equipment including 2 electric motors, etc., to Jones Hettelsater Constr. Co., 706 Mutual Bldg., Kansas City, Mo., at $10,000.

Okla., Guymon—The city has awarded the contract for power plant, etc., including one 75 k.w. and one 120 k.w. generators, two 150 hp. fire tube boilers, to Smith & Whitney, Dallas, Tex., at $78,305.

PRICES - MATERIALS - SUPPLIES

Electrical prices on following page are prices to the power plant by jobbers in the larger buying centers east of the Mississippi. Elsewhere the prices will be modified by increased freight charges and by local conditions.

POWER-PLANT SUPPLIES

HOSE—

Fire	50-Ft. Lengths
Underwriters' 2½-in.	63c. per ft.
Common, 2½-in.	50-10%

Air			
	First Grade	Second Grade	Third Grade
¾-in. per ft.	$0.40	$0.25	$0.19

Steam—Discounts from List

First grade	40%	Second grade	10-10-5%
		Third grade	60%

RUBBER BELTING—The following discounts from list apply to transmission rubber and duck belting:

Competition	60-10-5%	Best grade	40-10%
Standard	50-5%		

LEATHER BELTING—Present discounts from list in fair quantities (½ doz. rolls):

Light Grade	Medium Grade	Heavy Grade
45%	30-10-5%	30-5%

RAWHIDE LACING { For cut, best grade, 40-10%, 2nd grade, 45-10%. For laces in sides, best, 53c per sq. ft.; 2nd, 51c. Semi-tanned: cut, 30-10%; sides, 55c per sq. ft.

PACKING—Prices per pound:

Rubber and duck for low-pressure steam	$1.00
Asbestos for high-pressure steam	2.00
Duck and rubber for piston packing	1.00
Flax, regular	1.20
Flax, waterproofed	1.70
Compressed asbestos sheet	1.00
Wire insertion asbestos sheet	1.50
Rubber sheet	.45
Rubber sheet, wire insertion	.70
Rubber sheet, duck insertion	.50
Rubber sheet, cloth insertion	.30
Asbestos packing, twisted or braided and graphited, for valve stems and stuffing boxes	1.50
Asbestos wick, ½- and 1-lb. balls	.85

PIPE AND BOILER COVERING—Below are part of standard lists, with discounts.

PIPE COVERING		BLOCKS AND SHEETS	
Pipe Size	Standard List Per Lin.Ft.	Thickness	Price per Sq.Ft.
1-in.	$0.27	½-in.	$0.27
2-in.	.36	1 -in.	.30
6-in.	.80	1½-in.	.45
4-in.	.60	2 -in.	.60
3-in.	.45	2½-in.	.75
8-in.	1.10	3 -in.	.90
10-in.	1.30	3½-in.	1.05

85% magnesia high pressure ..... List +5%

For low-pressure heating and return lines { 4-ply ..... 50% off; 3-ply ..... 52% off; 2-ply ..... 54% off

PORTLAND CEMENT—New York, $3.10 without bags, in cargo lots delivered on job. Bag charge now 40c. per bbl.

STRUCTURAL STEEL—New York delivered price, 3 to 15-in. beams and channels and 3 to 6-in. angles, tees' and plates, all $3.23 per 100 lb.

COTTON WASTE—The following prices are in cents per pound:

	New York		
	Current	Cleveland	Chicago
White	10.00@13.00	13.00	14.25
Colored mixed	7.00@11.00	11.00	12.00

WIPING CLOTHS—Jobbers' price per 1000 is as follows:

	13½ x 13½	13½ x 20½
Cleveland	$55.00	$65.00
Chicago	41.00	43.50

LINSEED OIL—These prices are per gallon:

	New York	Cleveland	Chicago
Raw in barrels (5 bbl. lots)	$0.73	$0.90	$0.82
5 gal. cans	.76*	1.15	1.07

*Charge of $2.25 for two cans.

WHITE AND RED LEAD—Base price per pound:

	Red				White	
	Current		1 Year Ago		Current	1 Yr. Ago
	Dry	In Oil	Dry	In Oil	Dry and In Oil	Dry and In Oil
100-lb. keg	13.00	14.50	15.00	16.50	13.00	15.00
25- and 50-lb. kegs	13.25	14.75	15.25	16.75	13.25	15.25
12½-lb. keg	13.50	15.00	15.50	17.00	13.50	15.50
1-lb. cans	16.00	17.50	17.50	18.50	16.00	17.00
5-lb. cans	18.00	19.50	19.50	19.50	18.00	18.00

500 lb. lots less 10% discount; 2000 lb. lots less 10-2½%.

RIVETS—The following quotations are allowed for fair-sized orders from warehouse:

	New York	Cleveland	Chicago
Steel ½ and smaller	45%	50%	45%
Tinned	45%	50%	30%

Structural rivets, ¾, ⅞, 1 in. diameter by 2 in. to 5 in. sell as follows per 100 lb.:

New York	$5.08	Chicago	$5.08	Pittsburgh	$3.90

Boiler rivets, same sizes:

New York	$5.18	Chicago	$5.18	Pittsburgh	$4.00

REFRACTORIES—Prices in carlots:

Chrome brick, eastern Shipping points	net ton	75@ 80
Chrome cement, 40@45% Cr_2O_3	net ton	40@ 45
Chrome cement, 40@45% Cr_2O_3 in sacks	net ton	44@ 49
Clay brick, 1st quality, fire clay, 9-in. shapes, Pennsylvania, Ohio and Kentucky	per 1000	55@ 60
Clay brick, 2d quality, fire clay, 9-in. shapes Pennsylvania, Ohio and Kentucky	per 1000	45@ 50
Magnesite brick: 9-in. straight	net ton	90@ 100
Magnesite brick: 9-in. arches, wedges and keys	net ton	99@ 110
Magnesite brick: Soaps and splits	net ton	126@ 140
Silica brick: Chicago district	per 1000	55@ 60
Silica brick: Birmingham, Ala.	per 1000	46@ 51
Silica brick: Mt. Union, Pa.	per 1000	45@ 50

BABBITT METAL—Warehouse prices in cents per pound:

	New York	Cleveland	Chicago
Best grade	70.00	46.00	35.00
Commercial	30.00	16.50	9.00

COLD FINISHED STEEL—Warehouse prices are as follows:

	New York	Chicago	Cleveland
Round shafting or screw stock, per 100 lb. base	$5.50	$5.25	$4.84
Flats, square and hexagons, per 100 lb. base	6.00	5.75	5.34

BOILER SPECIALTIES—F. o. b. New York or Jersey City, discounts on list:

	Current
Copper ferrules	70%
Boiler flanges	50%
Boiler lugs	Net list
Boiler stay bolts	60%
Boiler patch bolts	+10%
Boiler fitting-up bolts	25%
Pressed steel boiler lugs	Net list
Pressed steel boiler hangers	Net list

WROUGHT PIPE—The following discounts are to jobbers for carload lots on the Pittsburgh basing card:

BUTT WELD

Steel			Iron		
Inches	Black	Galv.	Inches	Black	Galv.
¼ to 3	57½	44	¼ to 1½	24½ to 25½	8½ to 09½

LAP WELD

2	50½	38	2	20½ to 21	6½ to 07
2½ to 6	53½	41	2½ to 6	22½ to 23	9½ to 10
7 to 12	50½	37	7 to 12	19½ to 20	6½ to 07
13 and 14	41				
15	38½				

BUTT WELD, EXTRA STRONG, PLAIN ENDS

¼ to 1½	55½	43	¼ to 1½	24½ to 25½	9½ to 10½
2 to 3	56½	44			

LAP WELD, EXTRA STRONG, PLAIN ENDS

2	48½	37	2	21½ to 22½	8½ to 09
2½ to 4	51½	40	2½ to 4	23½ to 24	11½ to 12
4½ to 6	50½	39	4½ to 6	22½ to 23	10½ to 11
7 to 8	46½	33	7 to 8	14½ to 15	2½ to 03
9 to 12	41½	28	9 to 12	9½ to 10	5½ to+2½

BOILER TUBES—Following are prices in New York warehouse of tubes manufactured according to specifications of the American Society of Mechanical Engineers:

Size	Lapweld Steel	C. C. Iron	Seamless Steel
1	.....	.....	$0.27
1¼	.....	.....	.26
1½	.....	.....	.24
1¾	$0.2200	$0.3255	.26
2	.2100	.2834	.25
2¼	.2400	.3176	.29
2½	.2910	.3814	.36
2¾	.3298	.4323	.39
3	.3686	.4781	.43
3¼	.4365	.5663	.51
3½	.4032	.5728	.55
4	.5124	.70	.72

Tubes 2¼ in. diameter, or smaller, over 18 ft. long, 10 per cent extra.

These prices are net per lineal foot based on stock lengths. If cut to special lengths, billing will be based on the entire stock lengths.

In addition to the above, standard cutting charges are as follows:

1¼ in. to 2 in. diameter, 5c. per cut. 2¾ in. diameter, 7c. per cut
2½ in. diameter, 6c. per cut. 3 in. diameter, 9c. per cut
3½ in. to 4 in. diameter, 10c.

ELECTRICAL SUPPLIES

ARMORED CABLE—

B. & S. Size	Two Cond. ½ M Ft.	Three Cond. M Ft.	Two Cond. Lead M Ft.	Three Cond. Lead M Ft.
No. 14 solid	$ 65.00(net)	$138.00	$164.00	$210.00
No. 12 solid	135.00	170.00	225.00	265.00
No. 10 solid	185.00	235.00	275.00	325.00
No. 8 stranded	285.00	375.00	520.00	500.00
No. 6 stranded	400.00	500.00	560.00	.....

From the above lists discounts are:

Less than coil lots	10%	List
Coils to 1,000 ft.	—15%	10%
1,000 ft. and over	—20%	—15%

BATTERIES, DRY—Regular No. 6 size red seal, Columbia, or Ever-Ready:

	Each, Net
Less than 12	$0.43
12 to 50	.41
50 to 125 (bbl.)	.38
125 (bbl.) or over	.35

CONDUIT, Price per 1000 ft.; ELBOWS AND COUPLINGS, Per 100 pieces, f. o. b. New York, with 10-day discount of 5 per cent.

Size, In.	Conduit Black 2,500 to 5,000 Lbs.	Conduit Galvanized 2,500 to 5,000 Lbs.	Elbows Black 2,500 to 5,000 Lbs.	Elbows Galvanized 2,500 to 5,000 Lbs.	Couplings Black 2,500 to 5,000 Lbs.	Couplings Galvanized 2,500 to 5,000 Lbs.
¼	74 30	79.40	17.75	18.90	6.55	6.95
⅜	98.20	105.10	23.35	24.85	9.35	9.95
½	145.20	155.40	34.55	36.80	12.15	12.90
¾	196.40	210.20	44.25	46.95	16.75	17.75
1	234.85	251.35	59.00	62.65	20.65	21.90
1¼	316.00	338.20	108.25	114.85	27.55	29.20
1½	499.60	534.70	177.10	187.90	39.35	41.75
2	653.30	699.20	472.30	501.10	59.05	62.65
2½	813.30	886.50	1,043.05	1,106.65	78.70	83.50
3	995.35	1,050.75	1,205.40	1,278.90	98.40	104.40

CONDUIT NON-METALLIC, LOOM—

Size I. D., In.	Feet per Coil	List, Ft.
⅜	250	$0.05½
½	250	.06
⅝	250	.09
¾	200	.12
⅞	200	.15
1	150	.18
1	100	.25
1¼	100	.33
1½	Odd lengths	.40
2	Odd lengths	.55

1000 ft. and over.... 55%
Coils..... 45%
Less coils, 35%

CUT-OUTS—Following are net prices each in standard-package quantities:

CUT-OUTS, PLUG

S. P. M. L.	$0.12	D. P. D. B.	$0.43
D. P. M. L.	.21	T. P. to D. P. S. B.	.26
T. P. M. L.	.33	T. P. S. B.	.38
D. P. S. B.	.23	T. P. D. B.	.64

CUT-OUTS, N. E. C. FUSE

	0-30 Amp.	31-60 Amp.	60-100 Amp.
D. P. M. L.	$0.50	$1.30	$1.68
T. P. M. L.	.70	1.70	2.40
D. P. S. B.	.63	1.60	.....
T. P. S. B.	1.15	2.40	.....
D. P. D. B.	1.10	3.00	.....
T. P. D. B.	1.90	5.40	.....
T. P. to D. P. D. B.	1.30	3.80	.....

FLEXIBLE CORD—Price per 1,000 ft. in coils of 250 ft.:

No. 18 cotton twisted	$17 72
No. 16 cotton twisted	21.89
No. 18 cotton parallel	22.00
No. 16 cotton parallel	25.40
No. 18 cotton reinforced heavy	31.62
No. 16 cotton reinforced heavy	35.45
No. 18 cotton reinforced light	29.30
No. 16 cotton reinforced light	32.00
No. 18 cotton Canvasite cord	22.97
No. 16 cotton Canvasite cord	27.06

FUSES, ENCLOSED—

250-Volt	Std. Pkg.	List	600-Volt	Std. Pkg.	List
3-amp. to 30-amp.,	100	$0.25	3-amp. to 30-amp.,	100	$0.40
35-amp. to 60-amp.,	100	035	35-amp. to 60-amp.,	100	.60
65-amp. to 100-amp.,	50	.90	65-amp. to 100-amp.,	50	1.50
110-amp. to 200-amp.,	25	2.00	110-amp. to 200-amp.,	25	2.50
225-amp. to 400-amp.,	25	3.60	225-amp. to 400-amp.,	25	5.50
425-amp. to 600-amp.,	10	5.50	450-amp. to 600-amp.,	10	8.00

Discount: Less 1-5th standard package, 15%; 1-5th to standard pak age, 28%; tandard package 45%.

RENEWABLE FUSES, ENCLOSED—

Sizes	250-Volt List-Price	600-Volt List-Price	Std. Pkg. Quantity	Carton Quantity
1 to 30-amp.	$0.50 ea.	$1.10 ea.	100	10
35 to 60-amp.	1.00 ea.	1.25 ea.	100	10
65 to 100-amp.	2.00 ea.	3.00 ea.	50	5
110 to 200-amp.	4.00 ea.	5.00 ea.	25	5
225 to 400-amp.	7.50 ea.	11.00 ea.	25	1
450 to 600-amp.	11.00 ea.	16.00 ea.	10	1
450 to 600-amp.	11.00 ea.	16.00 ea.	10	1

RENEWAL LINE FOR ABOVE FUSES—

1 to 30	$0.30 ea.	$0.05	100	100
35 to 60	.05 ea.	.06	100	100
65 to 100	.10 ea.	.10	50	50
110 to 200	.15 ea.	.15	25	50
225 to 400	.30 ea.	.30	25	25
450 to 600	.60 ea.	.60	10	10

Discount Without Contract—Fuses:	
Broken carton	Net list
Unbroken carton but less than std. pkg.	10%
Std. pkg.	18%
Discount Without Contract—Renewals:	
Less than std. pkg.	Net list
Standard package	18%
Discount With Contract—Fuses:	
Broken cartons	5%
Unbroken cartons but less than standard package	15%
Standard package	23%
Discount With Contract—Renewals:	
Less standard package	Net list
Standard package	23%

FUSE PLUGS, MICA CAP—

0-30 ampere, standard package	$5.25C
0-30 ampere, less than standard package	6.00C

LAMPS—Below are present quotations in less than standard package quantities:

Straight-Side Bulbs Mazda B— Watts	Plain	Frosted	No. in Package	Pear-Shaped Bulbs Mazda C— Watts	Clear	Frosted	No. in Package
10	$0.40	$0.45	100	75	$0.75	$0.80	50
15	.40	.45	100	100	1.10	1.25	24
25	.40	.45	100	150	1.55	1.70	24
40	.40	.45	100	200	2.10	2.30	24
50	.40	.45	100	300	3.15	3.40	24
60	.45	.50	100				12
				500	4.60	4.90	12
				750	6.50	6.90	8
				1,000	7.50	7.95	8

Standard quantities are subject to discount of 10% from list. Annual contracts ranging from $150 to $300,000 net allow a discount of 17 to 40% from list.

PLUGS, ATTACHMENT—

	Each
Porcelain separable attachment plug	$0.28
Composition 2-piece attachment plug	.36
Swivel attachment plug	.20
Current taps	.45

RUBBER-COVERED COPPER WIRE—Per 1000 ft. f. o. b. New York.

No.	Solid Single Braid	Solid Double Braid	Stranded, Double Braid	Duplex
14	$ 8.25	$11.50	$15.08	$23.00
12	12.00	15.94	18.47	30.56
10	10.59	19.49	23.42	39.19
8	22.82	26.77	30.91	52.60
6	.....	.....	42.70	.....
4	.....	.....	59.00	.....
2	.....	.....	85.13	.....
1	.....	.....	107.95	.....
0	.....	.....	127.62	.....
00	.....	.....	150.14	.....
000	.....	.....	182.57	.....
0000	.....	.....	224.67	.....

SOCKETS, BRASS SHELL—

¼ In. or Pendant Cap Key Each	Keyless Each	Pull Each	⅛ In. Cap Key Each	Keyless Each	Pull Each
$0.33	$0.30	$0.60	$0.39	$0.36	$0.66

Less 1-5th standard package	+50%
1-5th to standard package	+20%
Standard package	—15%

WIRE, ANNUNCIATOR AND DAMPPROOF OFFICE—

No. 18 B. & S. regular spools (approx. 8 lb.)	47c. lb.
No. 18 B. & S. regular 1-lb. coils	48c. lb.

WIRING SUPPLIES—

Friction tape, ¾ in., less 100 lb. 56c. lb., 100 lb. lots	52c. lb.
Rubber tape, ¾ in., less 100 lb. 65c. lb., 100 lb. lots	60c. lb.
Wire solder, less 100 lb. 47c. lb., 100 lb. lots	42c. lb.
Soldering paste, 2 oz. cans	$1.50 doz.

SWITCHES, KNIFE—

TYPE "C" NOT FUSIBLE

Size, Amp.	Single Pole, Each	Double Pole, Each	Three Pole, Each	Four Pole, Each
30	$0.42	$0.68	$1.02	$1.36
60	.74	1.22	1.84	2.44
100	1.50	2.50	3.76	5.00
200	2.70	4.50	6.76	9.00

TYPE "C" FUSIBLE, TOP OR BOTTOM

30	.70	1.06	1 60	2.12
60	1.18	1.80	2.70	3.60
100	2.38	3.66	5.50	7.30
200	4.40	6.76	10.14	13.50

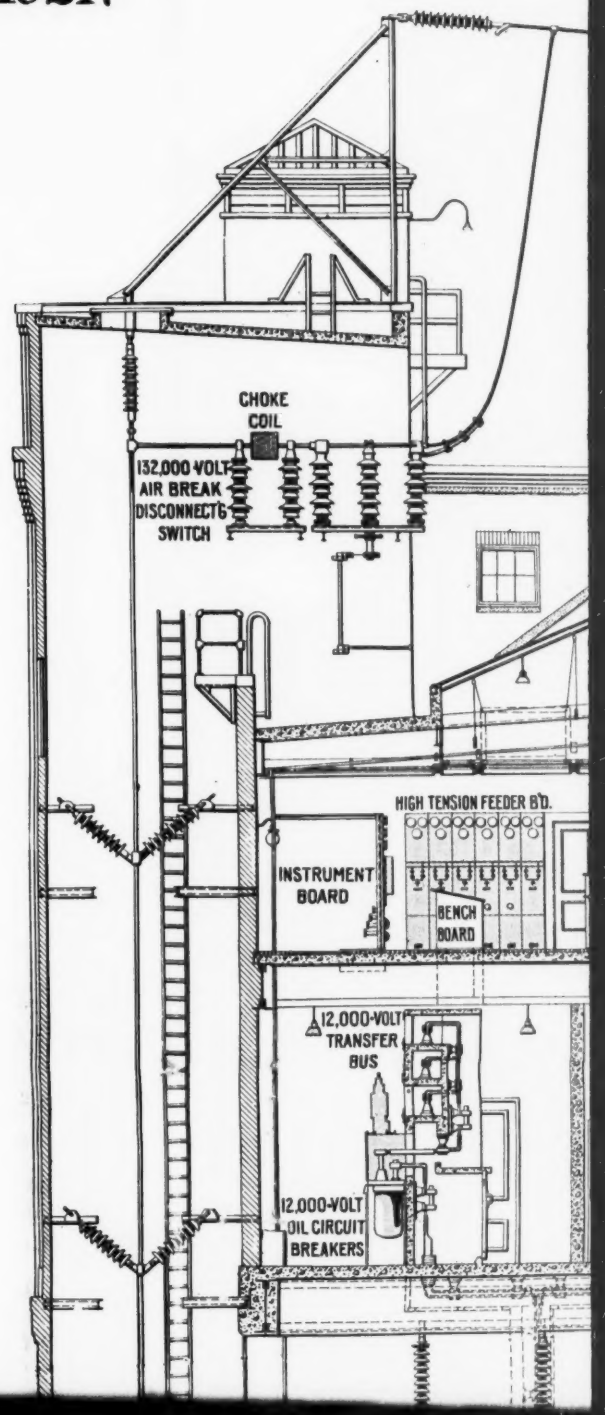
Discounts:	
Less than $10.00 list value	+25%
$10 to $25 list value	+10%
$25 to $50 list value	+ 5%
$50 to $200 list value	—10%
$200 list value or over	—15%

Cross-Section Colfax Power Sta

Duquesne Light Company

Pittsburgh, Pa.

Station planned for six 60,000-kw. units. One machine in op
and second unit on order, April 5, 1921.



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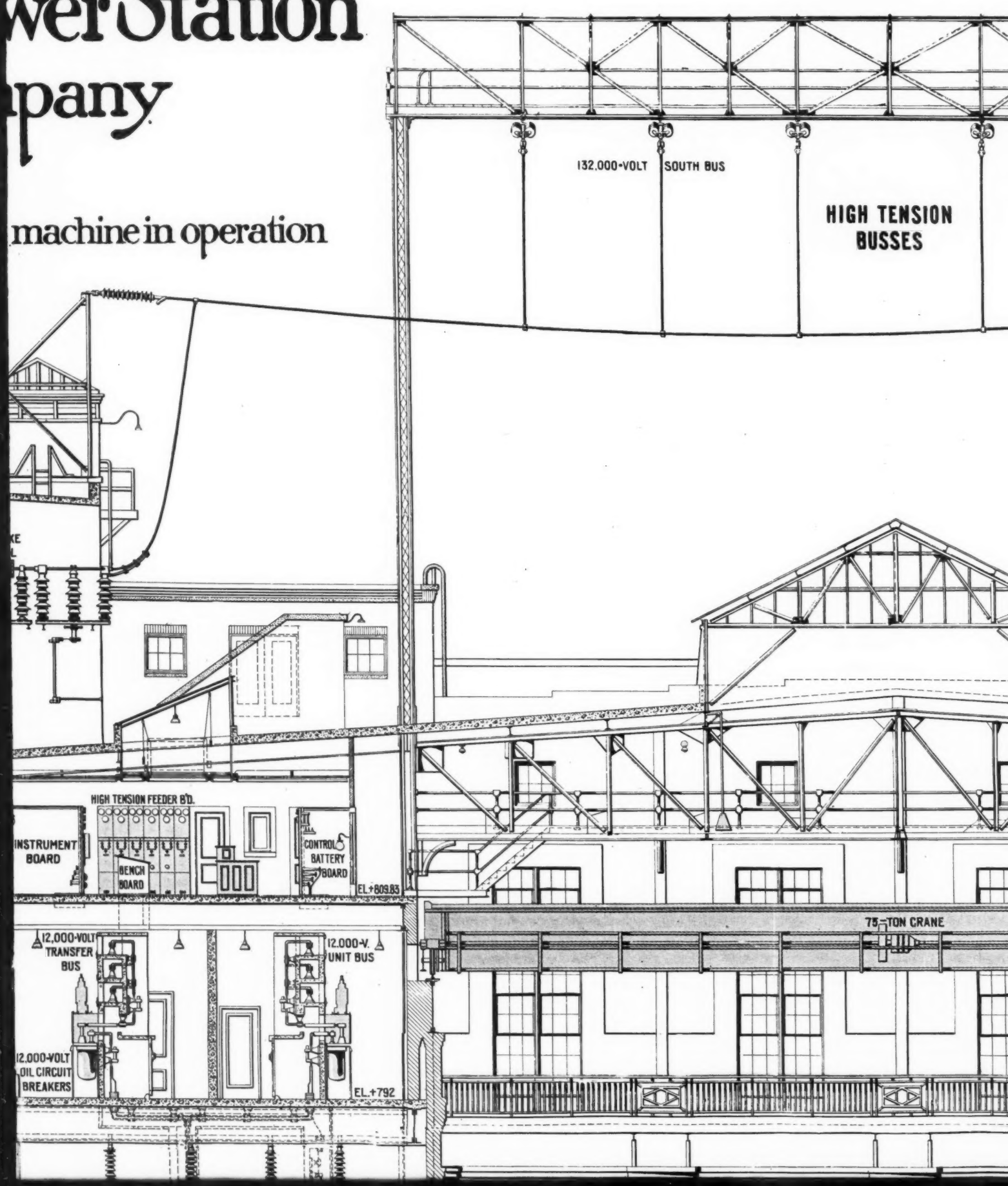
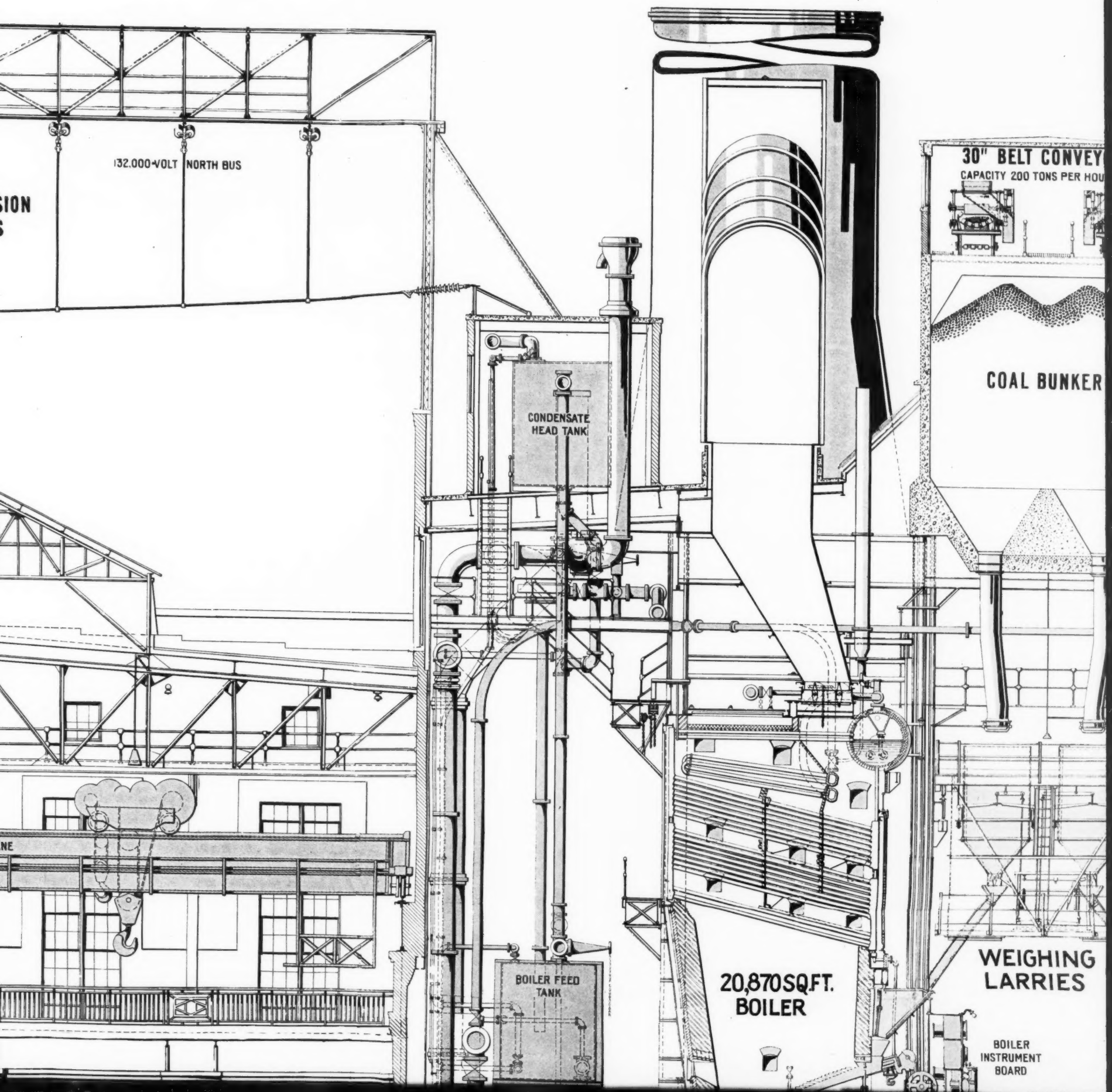
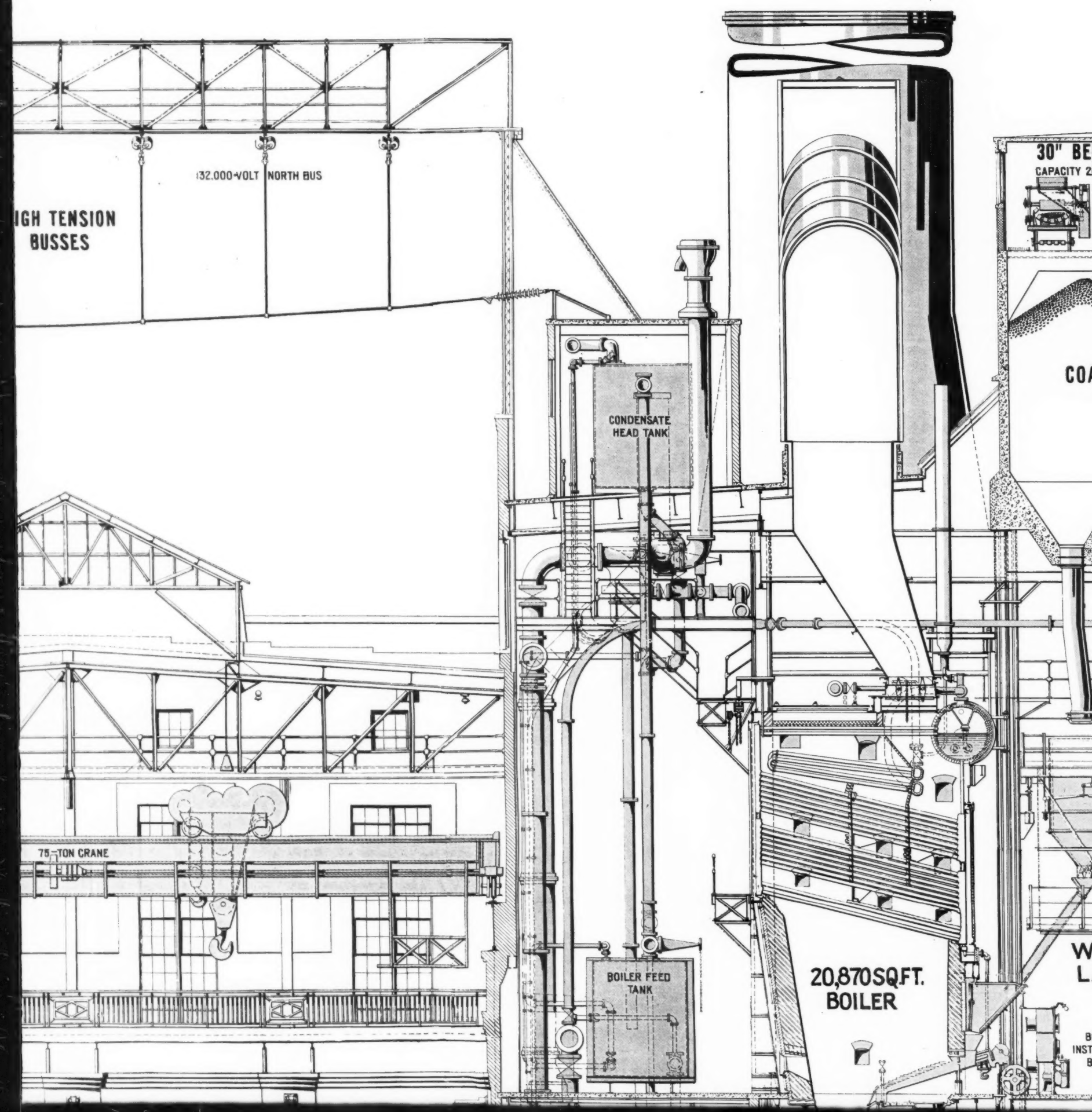
machine in operation




132,000-VOLT NORTH BUS
CONDENSATE HEAD TANK
30" BELT CONVEY
CAPACITY 200 TONS PER HOU
COAL BUNKER
BOILER FEED TANK
20,870 SQ.FT. BOILER
WEIGHING LARRIES
BOILER INSTRUMENT BOARD


132,000-VOLT NORTH BUS
IGH TENSION
BUSSES
CONDENSATE
HEAD TANK
30" BE
CAPACITY 2
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75-TON CRANE
BOILER FEED
TANK
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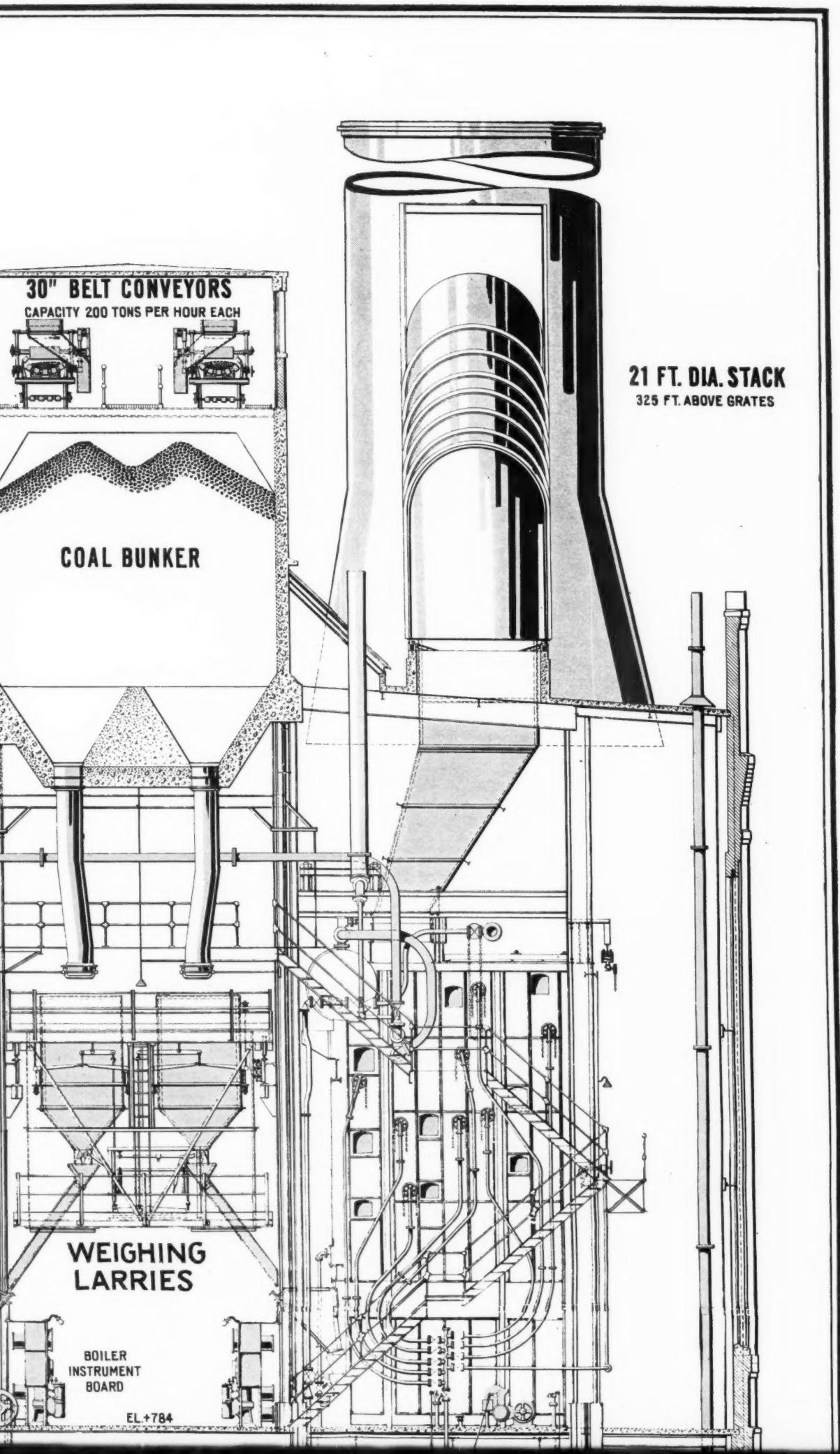
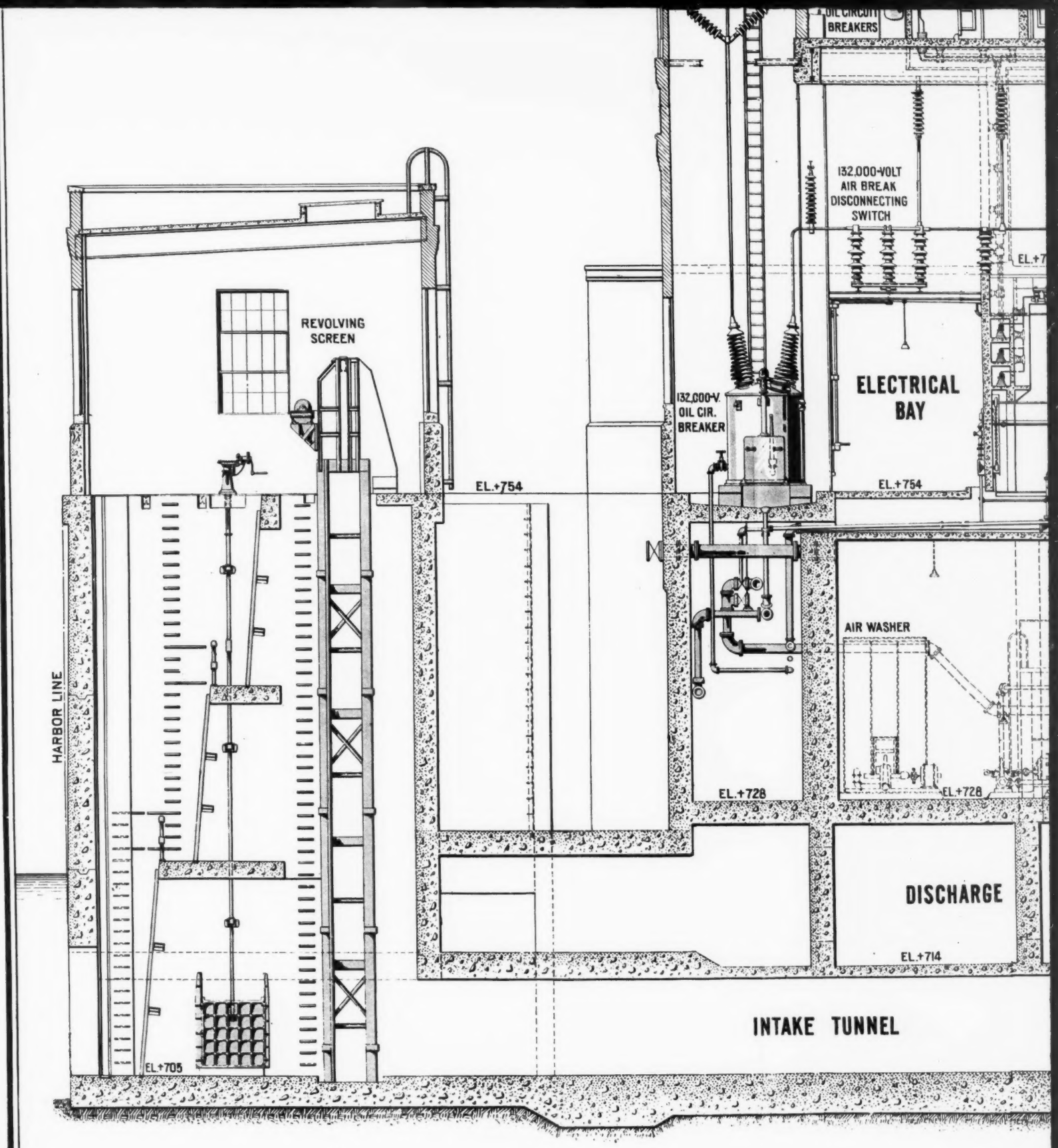

30" BELT CONVEYORS
CAPACITY 200 TONS PER HOUR EACH
21 FT. DIA. STACK
325 FT. ABOVE GRATES
COAL BUNKER
WEIGHING LARRIES
BOILER INSTRUMENT BOARD
EL.+784


OIL CIRCUIT BREAKERS
132,000-VOLT AIR BREAK DISCONNECTING SWITCH
EL.+7
REVOLVING SCREEN
132,000-V. OIL CIR. BREAKER
ELECTRICAL BAY
EL.+754
EL.+754
AIR WASHER
HARBOR LINE
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DISCHARGE
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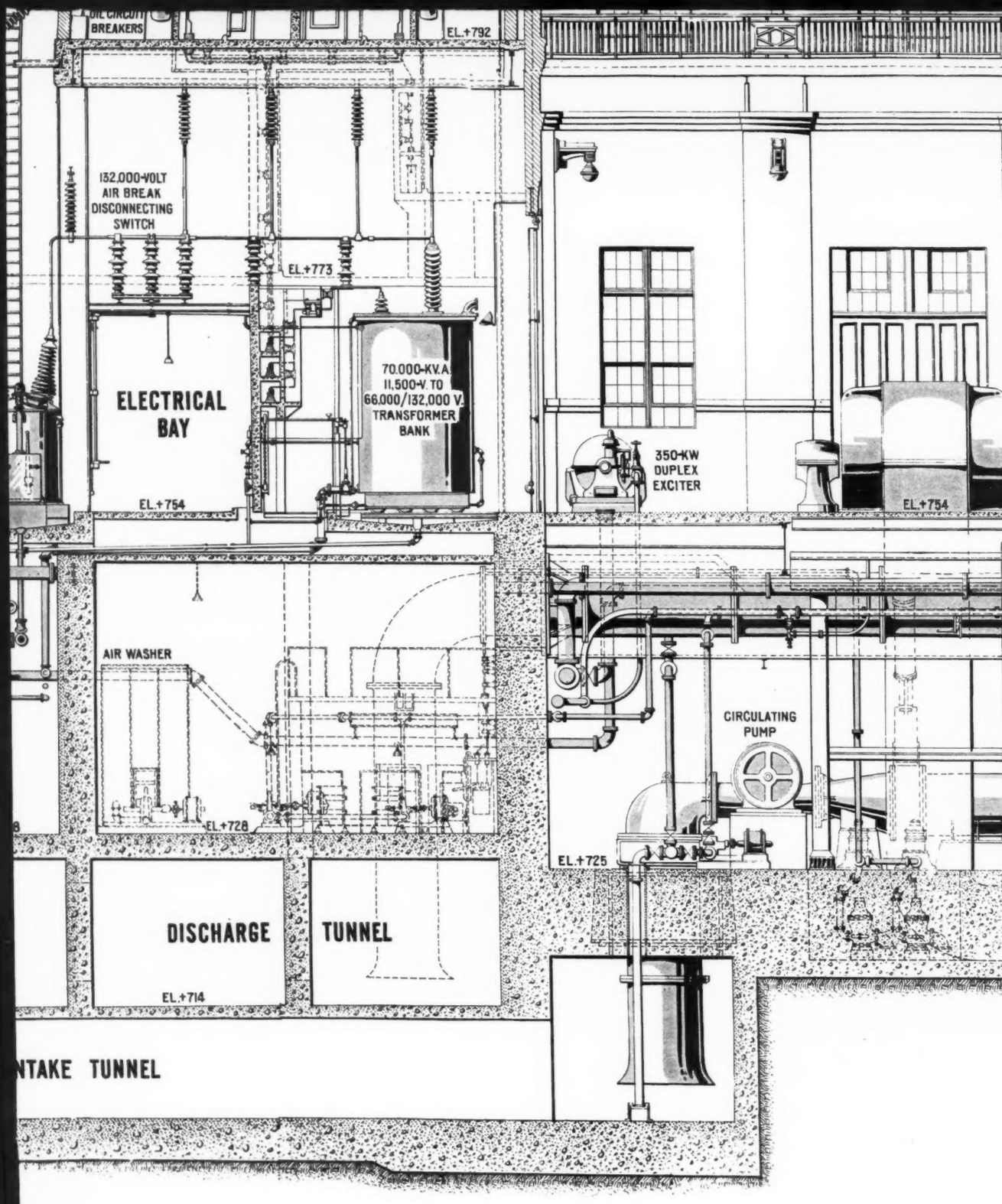
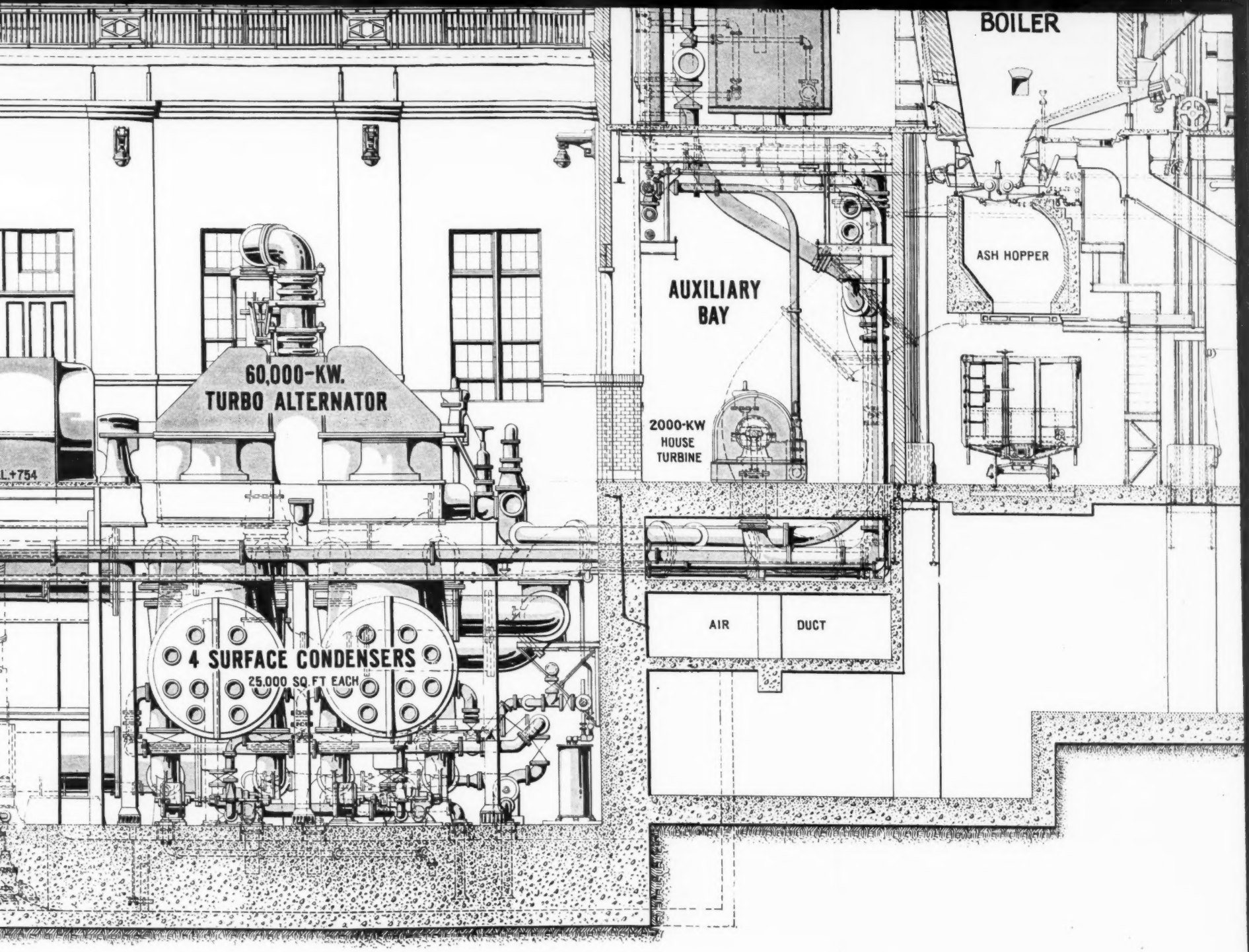
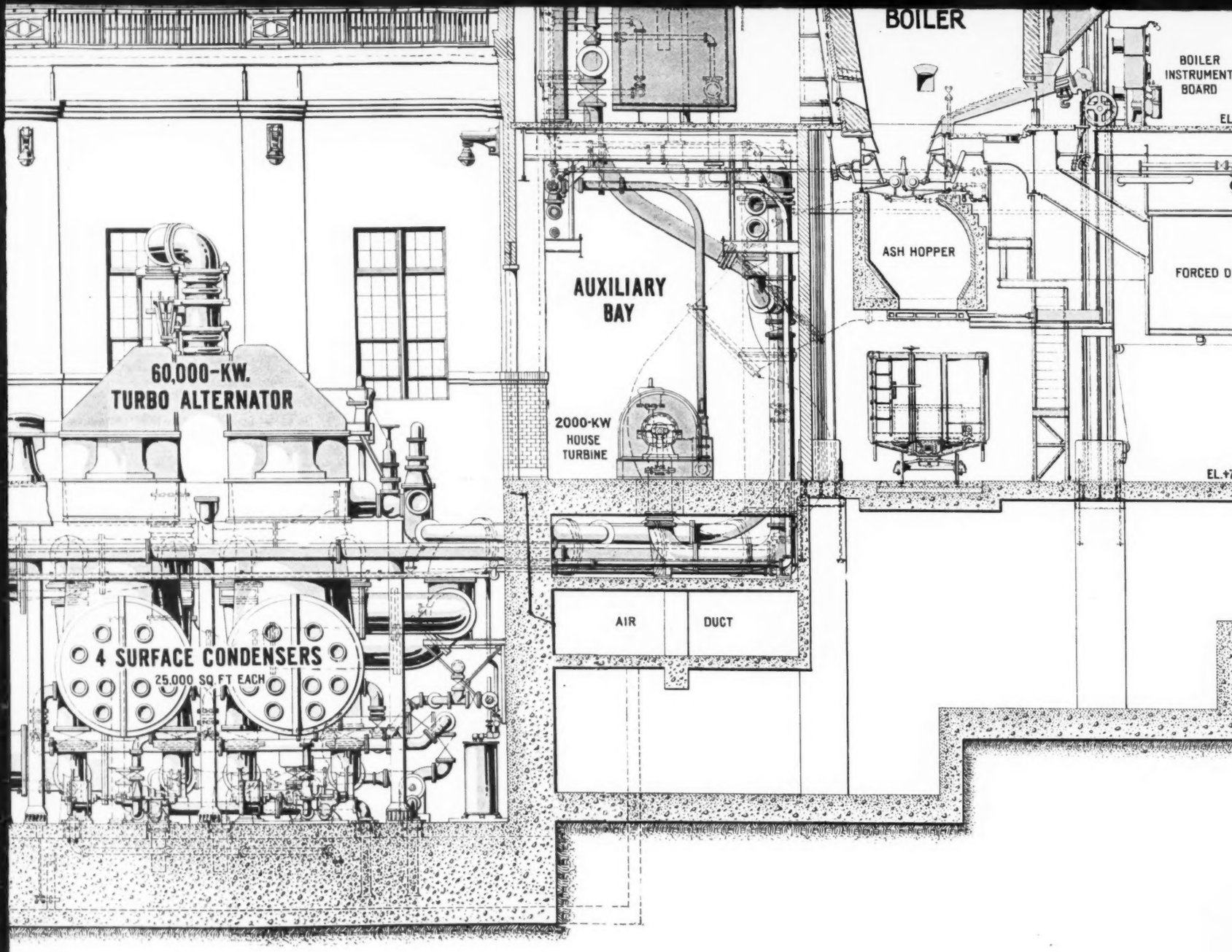
INTAKE TUNNEL
EL.+705

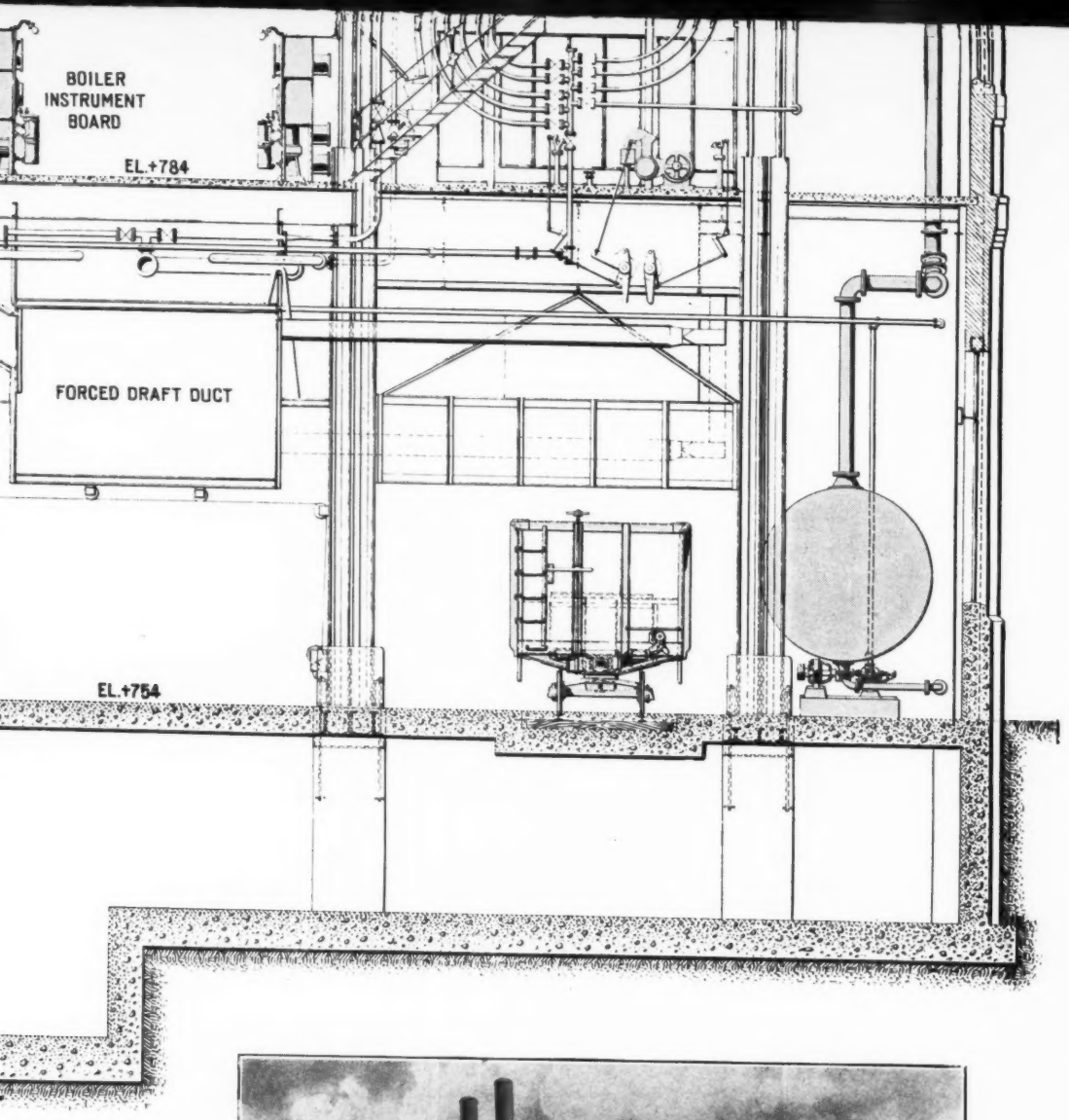

OIL CIRCUIT BREAKERS
EL.+792
132,000-VOLT AIR BREAK DISCONNECTING SWITCH
EL.+773
70,000-KV.A. 11,500-V. TO 66,000/132,000 V. TRANSFORMER BANK
ELECTRICAL BAY
EL.+754
350-KW DUPLEX EXCITER
EL.+754
AIR WASHER
EL.+728
CIRCULATING PUMP
EL.+725
DISCHARGE
TUNNEL
EL.+714
NTAKE TUNNEL

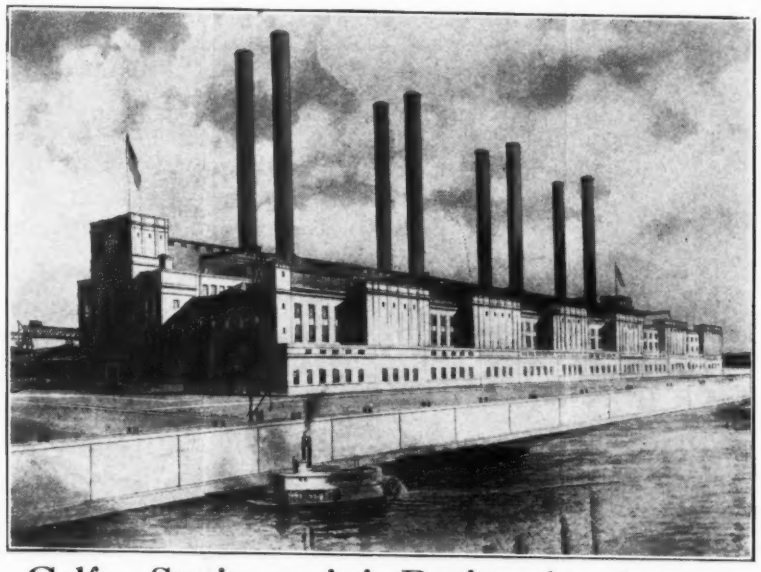


Information furnished by Dwight P. Ro



Information furnished by Dwight P. Robinson





Colfax Station as it is Designed to Appear
When Completed

obinson & Co. Inc., Engineers & Constructors, New York City.